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मानक

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IS/ISO 13373-1 (2002): Condition Monitoring and Diagnostics of Machines - Vibration Condition Monitoring, Part 1: General Procedures [MED 28: Mechanical Vibration and Shock]



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Bhartrhari—Nitiśatakam

“Knowledge is such a treasure which cannot be stolen”

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भारतीय मानक

मशीनों की अवस्था मॉनीटरिंग तथा डाइग्नोस्टिक —
कंपन स्थिति की मॉनीटरिंग

भाग 1 सामान्य विधियाँ

Indian Standard

CONDITION MONITORING AND DIAGNOSTICS OF
MACHINES — VIBRATION CONDITION MONITORING

PART 1 GENERAL PROCEDURES

ICS 17.160

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BUREAU OF INDIAN STANDARDS

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NATIONAL FOREWORD

This Indian Standard (Part 1) which is identical with ISO 13373-1 : 2002 'Condition monitoring and diagnostics of machines — Vibration condition monitoring — Part 1: General procedures' issued by the International Organization for Standardization (ISO) was adopted by the Bureau of Indian Standards on the recommendation of the Mechanical Vibration and Shock Sectional Committee and approval of the Mechanical Engineering Division Council.

The text of ISO Standard has been approved as suitable for publication as an Indian Standard without deviations. Certain conventions are, however, not identical to those used in Indian Standards. Attention is particularly drawn to the following:

- a) Wherever the words 'International Standard' appear referring to this standard, they should be read as 'Indian Standard'.
- b) Comma (,) has been used as a decimal marker, while in Indian Standards, the current practice is to use a point (.) as the decimal marker.

In this adopted standard, reference appears to certain International Standards for which Indian Standards also exist. The corresponding Indian Standards which are to be substituted in their respective places, are listed below along with their degree of equivalence for the editions indicated:

<i>International Standard</i>	<i>Corresponding Indian Standard</i>	<i>Degree of Equivalence</i>
ISO 1925 : 2001 Mechanical vibration — Balancing — Vocabulary	IS/ISO 1925 : 2001 Mechanical vibration — Balancing — Vocabulary	Identical
ISO 2041 : 1990 Vibration and shock — Vocabulary	IS 11717 : 2000 Vocabulary on vibration and shock (<i>first revision</i>)	do
ISO 7919-1 : 1996 Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria — Part 1: General guidelines	IS 14773 (Part 1) : 2000 Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria: Part 1 General guidelines	do
ISO 10816-1 : 1995 Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 1: General guidelines	IS 14817 (Part 1) : 2000 Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts: Part 1 General guidelines	do

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2 : 1960 'Rules for rounding off numerical values (*revised*)'. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

*Indian Standard***CONDITION MONITORING AND DIAGNOSTICS OF
MACHINES — VIBRATION CONDITION MONITORING****PART 1 GENERAL PROCEDURES****1 Scope**

This part of ISO 13373 provides general guidelines for the measurement and data collection functions of machinery vibration for condition monitoring. It is intended to promote consistency of measurement procedures and practices, which usually concentrate on rotating machines.

Because of the diversity of approaches to condition monitoring, recommendations specific to a particular kind of monitoring programme will be addressed in additional parts of ISO 13373.

This part of ISO 13373 is a basic document which presents recommendations of a general nature, encompassing

- measurement methods,
- measurement parameters,
- transducer selection,
- transducer location,
- transducer attachment,
- data collection,
- machine operating conditions,
- vibration monitoring systems,
- signal conditioning systems,
- interfaces with data-processing systems,
- continuous monitoring, and
- periodic monitoring.

The vibratory conditions of a machine can be monitored by vibration measurements on the bearing or housing structure and/or by vibration measurements of the rotating elements of the machine. In addition, measurements can be continuous or non-continuous. This part of ISO 13373 provides guidance on the types of measurements recommended in both the continuous and the non-continuous modes.

It is emphasized that this part of ISO 13373 addresses only the procedures for vibration condition monitoring of machines. In many cases, the complete condition monitoring and diagnostics of a machine can also include other parameters, such as thermography, oil analysis, ferrography, process variations, temperatures and pressures. These non-vibratory parameters will be included in other International Standards.

This part of ISO 13373 covers rotating machines. However, many of the procedures included can be applied to other types of machines, for example reciprocating machines.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 13373. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO 13373 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 1925, *Mechanical vibration — Balancing — Vocabulary*

ISO 2041, *Vibration and shock — Vocabulary*

ISO 7919-1, *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria — Part 1: General guidelines*

ISO 10816-1, *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 1: General guidelines*

3 Terms and definitions

For the purposes of this part of ISO 13373, the terms and definitions given in ISO 1925 and ISO 2041 apply.

4 Vibration condition monitoring

4.1 General

Vibration monitoring is conducted to assist in the evaluation of the “health” of the machine during sustained operation. Depending on the machine type and the critical components to be monitored, one or more measurement parameters, and a suitable monitoring system, have to be selected. The objective of such a programme is to recognize an “unhealthy” condition in sufficient time to take remedial action before certain defects in the machine parts significantly decrease equipment operation or projected machine life, or fail completely, thereby establishing a cost-effective maintenance plan.

Several types of condition monitoring systems are described below; depending on the machine, the machine’s condition and other factors, any one of the systems, or combinations thereof, may be selected.

4.2 Types of vibration condition monitoring systems

4.2.1 General

Condition monitoring systems take many forms. They utilize permanently installed, semi-permanent or portable measuring equipment.

A decision to select the appropriate measuring system depends upon a number of factors, such as

- criticality of the machine operation,
- cost of machine down-time,
- cost of catastrophic failure,
- cost of the machine,

- rate of progress of the failure mode,
- accessibility for repair/maintenance (e.g. in nuclear plants or other remote locations),
- accessibility of the appropriate measurement positions,
- quality of the measurement/diagnostic system,
- operational modes of the machine (e.g. speed, power),
- cost of the monitoring system,
- safety, and
- environmental impacts.

4.2.2 Permanently installed systems

This type of system is one in which the transducers, signal conditioners, data-processing and data-storage equipment are permanently installed. Data may be collected either continuously or periodically. The application of permanently installed systems is usually limited to costly and critical machinery or to machines with complex monitoring tasks. Figure 1 shows a typical permanently installed on-line system.

4.2.3 Semi-permanent systems

The semi-permanent system is a cross between the permanent and portable systems. In this type of system the transducers are generally permanently installed, whereas the electronic data-acquisition components are intermittently connected.

4.2.4 Portable monitoring systems

A portable monitoring system performs similar functions as the “continuous” on-line system, but it is less elaborate and normally less expensive. With this arrangement, the data are recorded periodically either automatically or manually, with a portable data collector. This type of system is shown in Figure 2.

More commonly, portable monitoring systems are used to record manually measurements at preselected locations on the machine at periodic intervals (weekly, monthly, etc.). The data are usually entered and stored locally on a portable data collector. A preliminary cursory analysis can be done immediately; however, for more in-depth processing and analysis, the data is downloaded to a personal computer that has the appropriate software.

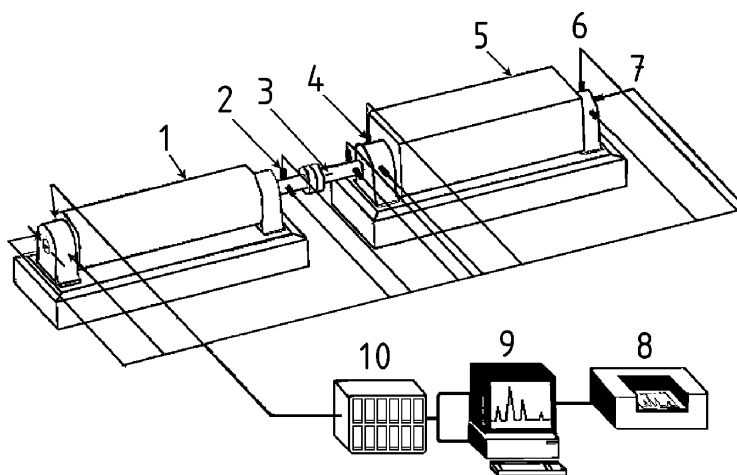
4.3 Data collection

4.3.1 General

Data may be collected on a continuous or periodic basis; and the data analysis may be driven by events or by intervals.

4.3.2 Continuous data collection

A continuous data-collection system is one in which vibration transducers are installed permanently at key locations on the machine (as shown in Figure 1), and in which the vibration measurements are usually recorded and stored continuously, during operation of the machine. It can include automatic vibration monitoring systems with multiplex connections provided that the multiplexing rate is sufficiently rapid so that no significant data or trends are lost. The data may be processed to provide either broadband or spectral information which can be compared to previously acquired data. By setting “alert limits” on the stored data it is possible to inform the operator that the vibratory pattern of the machine has changed (the magnitude has either increased or decreased), and therefore diagnostic procedures are recommended.

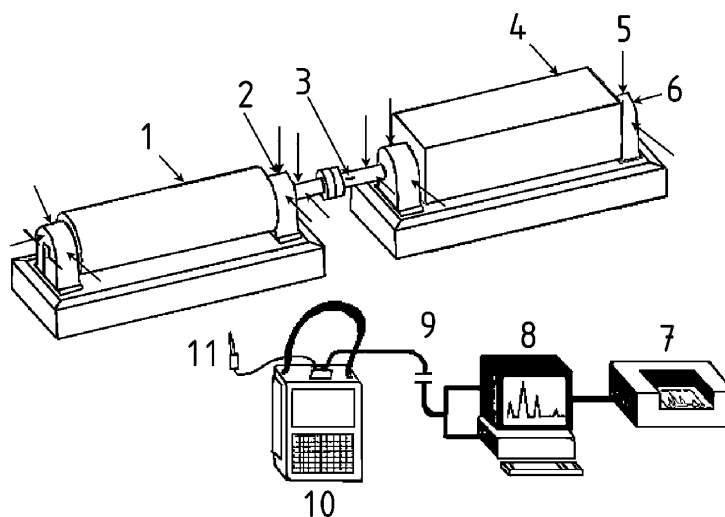


Key

- | | |
|---|------------------------------|
| 1 Driver | 6 Radial |
| 2 Shaft displacement probes (typical) | 7 Axial |
| 3 Phase reference | 8 Printer |
| 4 Transducers on stationary bearing structure (typical) | 9 Computer with data storage |
| 5 Driven process machinery | 10 Signal conditioner |

NOTE This figure shows one typical arrangement. Alternative systems are permissible (e.g. microprocessor-based systems often have integral signal conditioning which may be carried out after the A/D conversion).

Figure 1 — Typical permanently installed on-line vibration condition monitoring system



Key

- | | |
|----------------------------|------------------------------|
| 1 Driver | 7 Printer |
| 2 Data points (typical) | 8 Computer with data storage |
| 3 Phase reference | 9 Computer link |
| 4 Driven process machinery | 10 Portable data logger |
| 5 Radial | 11 Transducer |
| 6 Axial | |

Figure 2 — Typical portable monitoring system

A continuous data-collection system can be installed at the machine site for direct use by the machine operating crew, or it can be installed at remote sites with data transmitted to a central data analysis centre. The obvious advantage of a “continuous” system is the availability of on-line machine condition in real time.

In an automatic system, permanent vibration transducers are installed on the machine in much the same manner as with the continuous monitoring system. The system is programmed to record and store data automatically. The last data are compared with the previously stored data in order to determine whether an ALARM condition exists.

A decision to select a continuous data-collection system should be taken after consideration of the factors listed in 4.2.1.

4.3.3 Periodic data collection

Periodic data collection can be done with either permanent on-line or portable systems. On-line periodic systems may include automatic vibration monitoring systems with multiplexer connections. In this case all channels are cyclically scanned one after the other with respect to off-limit conditions. The measuring system is permanently in action but there are gaps in monitoring the individual measuring points, which are dependent on the number of channels monitored and the measuring period per channel. These systems are sometimes referred to as “scanning” or “intermittent” systems.

For machines for which permanent on-line systems cannot be justified, portable systems are usually used and they are in most cases suitable for periodic monitoring.

4.4 Condition monitoring programme

After selection of the equipment to be monitored, and determining the type of measurement system that is appropriate, it is recommended that a condition monitoring programme flowchart be developed. Figure 3 shows such a typical flowchart. However, since each plant and each system is unique, the data flow should be customized to provide the maximum benefits.

Clear descriptions of operating conditions, such as speed, load or temperature, should accompany any vibration data collected. As a minimum, such descriptions should include shaft speed (r/min) and machine load (power, flow, pressure, etc.) and any other operating parameter that can affect the measured vibration.

In general, during data acquisition it is strongly emphasized that the operating conditions should approximate the normal operating conditions of the machine as closely as possible, to ensure consistency and valid comparability of the data. When this is not possible, the characteristics of the machine must be well known in order to evaluate any differences in the data.

Since the procedure of condition monitoring includes the process of “trending”, which examines the rate at which vibration values change with operating time, it is especially important that the operating conditions during successive measurements remain the same, in order for such trending to be valid. For example, in the case of pumps, the vibration values can vary significantly between “normal” and “off-normal” operating loads. Thus, a change in vibration response due to a change in operating conditions could easily be incorrectly interpreted as a change due to an impending problem.

In addition, the time rate of data acquisition need not to be constant. As pointed out in 7.3, it depends on the current condition of the machine.

Data under other conditions may also need to be collected depending on the complexity of the machine and the purpose of the measurement. For example, where problems with unbalance, rubbing, shaft cracks or oil whirl are suspected, testing during transient operating conditions such as start-up and shut-down is recommended.

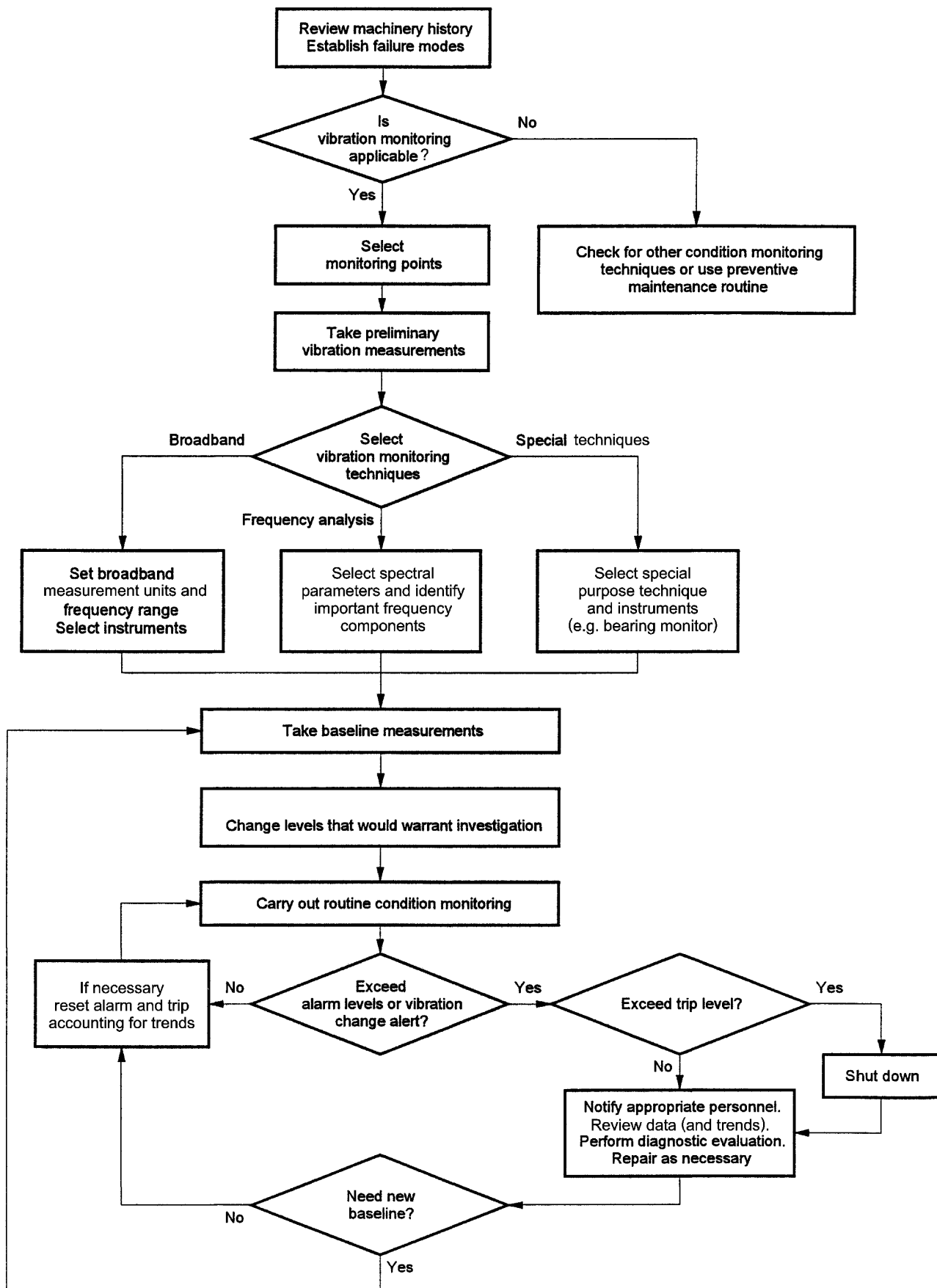


Figure 3 — Vibration condition monitoring flowchart

5 Measurements

5.1 General

This clause provides information on the types of measurements and the measurement quantities recommended for vibration condition monitoring. Annex B lists the typically required information to be recorded for each machine and measurement.

5.2 Types of measurements

5.2.1 General considerations

In general, there are three types of measurements that may be employed for machine vibration condition monitoring, as follows:

- a) vibration measurements made on the non-rotating structure of the machine, such as the bearing housings, machine casing or machine base;
- b) relative motion between the rotor and the stationary bearings or housing;
- c) absolute vibratory motion of the rotating elements.

Vibration measurements on structures would normally use r.m.s. velocity often combined with r.m.s. displacement or acceleration (see ISO 10816-1). If the vibration is predominantly sinusoidal, vibratory displacement (zero-to-peak or peak-to-peak) may also be used.

For high-speed machines/gears and machines with anti-friction bearings, peak acceleration is often used for monitoring in combination with r.m.s. velocity. In addition, increasing use is being made of other more sophisticated techniques which enable greater use to be made of the information contained within the vibration signal.

Absolute and relative displacement of the rotating components are further defined by several different displacement quantities, each of which is now in widespread use and is defined in ISO 7919-1. These include

- $S_{\max}$, the maximum value of shaft displacement from the time-integrated zero mean position, and
- S_{p-p} , the vibratory peak-to-peak displacement in the direction of measurement.

Each of these displacement quantities may be used for measurement of shaft vibration. However, the quantities shall be clearly identified so as to ensure correct interpretation of the measurements.

It should be noted that the ISO 7919 and ISO 10816 series address broadband measurements only. However, condition monitoring may include additional vibration measurements and analyses such as

- spectral analysis,
- filtering,
- time wave forms and orbits,
- vector analysis with magnitude and phase, and
- analysis of the high-frequency vibration envelope.

5.2.2 Transducer locations

The locations of transducers for the purposes of condition monitoring depend on the particular machine and the specific parameters to be measured. Before specifying "location", it is first necessary to identify which parameters should be monitored, i.e.

- the absolute vibration of the machine housing,
- the vibratory motion of the rotor relative to the housing,
- the position of the shaft relative to the housing during machine operation, and
- the absolute motion of the shaft.

In general, the transducers should be located at or near the bearings. However, if experience is available for a particular type of machine and if such transducer locations are practical, it could be useful to locate transducers at other positions than the bearings, as follows:

- a) at the positions which are most likely to provide maximum values of vibration (such as intermediate shaft mid-span of large gas turbine units);
- b) at the positions where a small clearance exists between the static and rotating parts and rubbing may occur.

Whatever the plane chosen for the vibration measurement, transducers should be located at those angular positions which are most likely to provide early indications of wear or failure.

5.2.3 Transducer locations for typical machines

Annex A includes a table showing recommended locations for obtaining meaningful vibration data for various types of machines. These locations and directions provide for shaft measurements near the bearings.

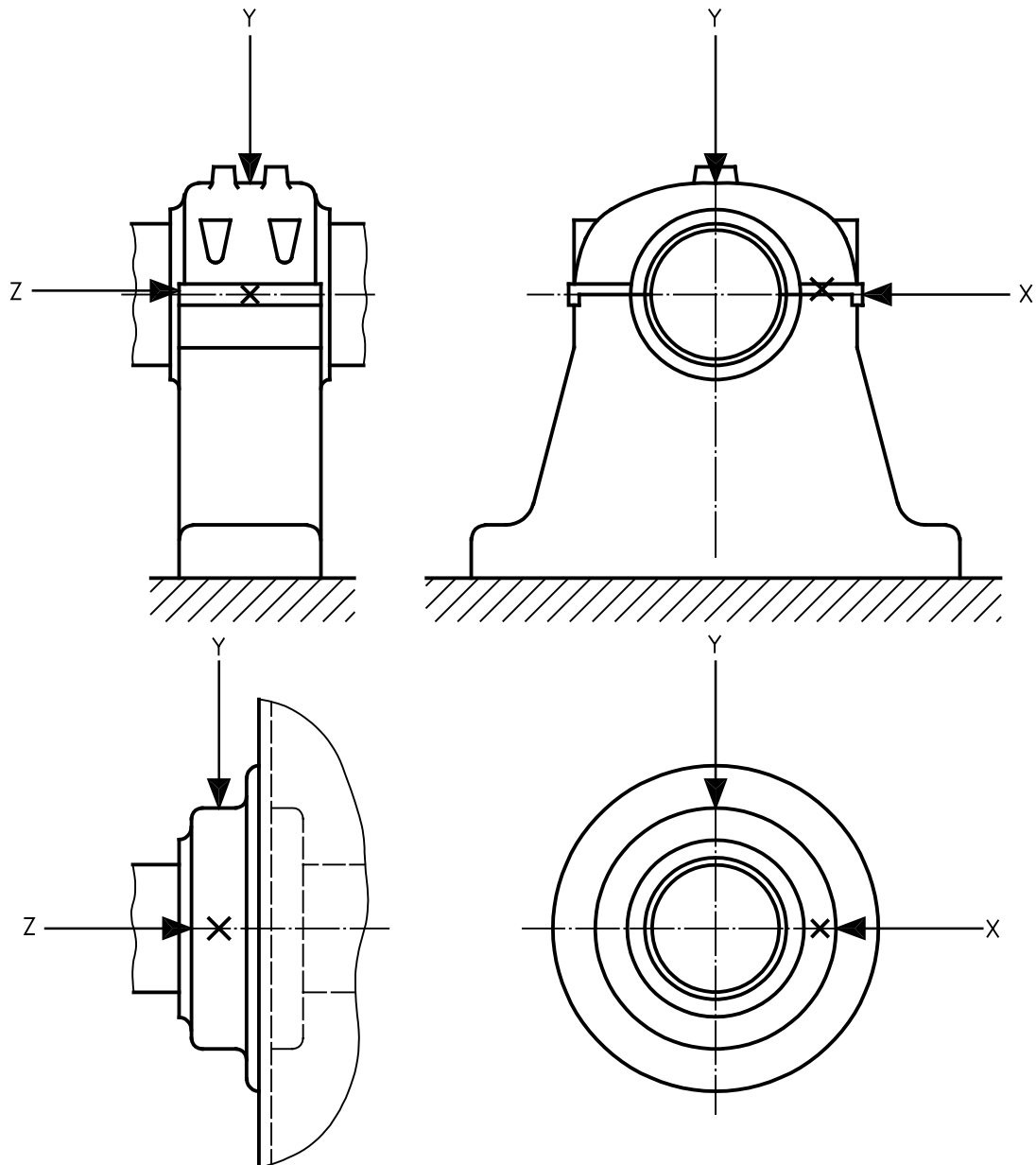
5.2.4 Transducer orientation and identification

The locations of the transducers should be clearly marked and identified to ensure repeatability of location during successive measurements. It is important to establish a consistent naming convention for assets and measurement points. Annex D provides an example of a typical convention for transducer orientation and identification.

5.2.5 Measurements on non-rotating structures

The ISO 10816 series provides specific procedures and instrumentation for different classes of machinery and identifies r.m.s. vibration velocity (in millimetres per second) as the preferred evaluation parameter. Measurements may be made either directly by a velocity transducer or by an accelerometer with an integrating circuit.

Typical locations for these types of measurements for horizontal machines are on each bearing housing or pedestal as shown in Figure 4. For vertical machines, see Figure 5.



X horizontal
Y vertical
Z axial

Figure 4 — Schematic diagram of typical transducer locations for vibration measurements on horizontal machines

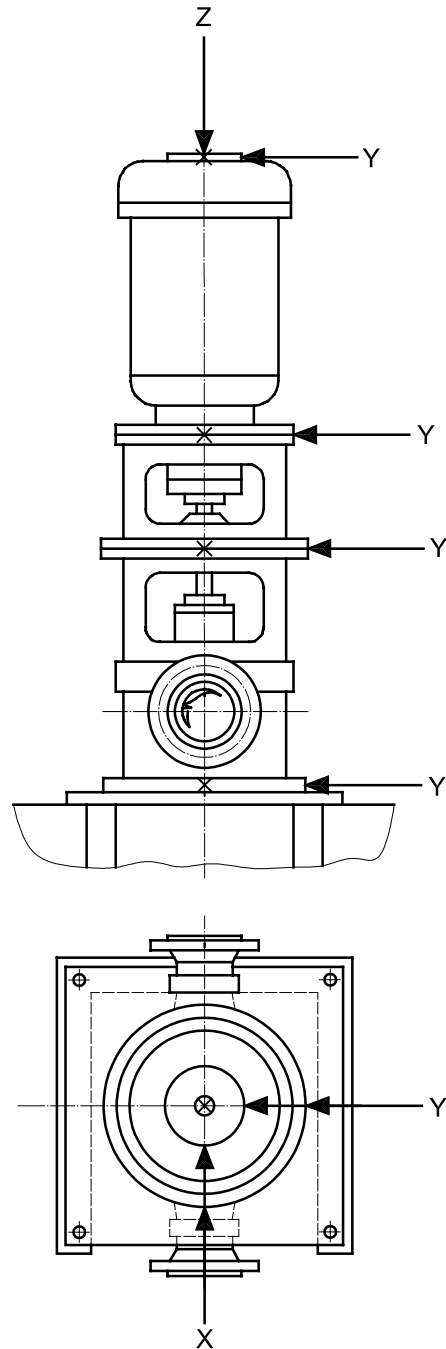


Figure 5 — Schematic diagram of typical transducer locations for vibration measurement on vertical machines

5.2.6 Measurements on rotating shafts

5.2.6.1 General

As defined in ISO 7919-1, the preferred measurement quantity for the measurement of shaft vibration is displacement. The following subclauses describe various transducers and arrangements for shaft measurements. Also shown on the figures are references to local signal conditioning and data processing. A discussion of signal conditioning is given in 5.2.8.

NOTE Details of further signal processing and analysis and displaying of the data will be given in ISO 13373-2 which is at present under preparation.

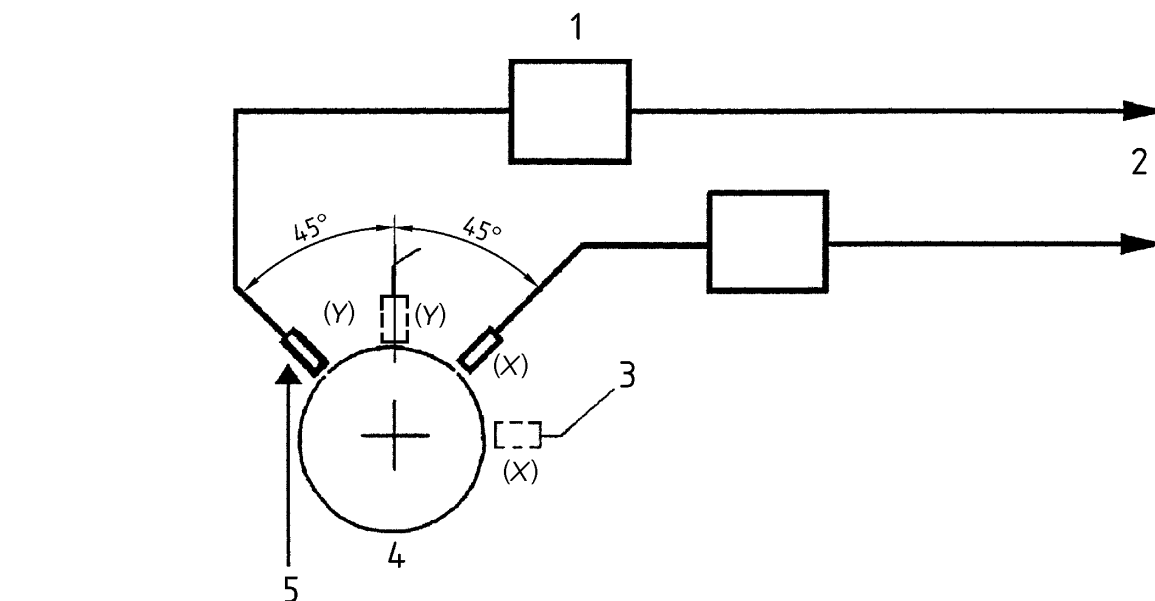
5.2.6.2 Relative motion of shaft to housing

The ISO 7919 series provides specific procedures and instrumentation for the measurement of broadband vibratory displacements of rotating shafts for different classes of machinery.

The relative displacement is best measured by using two non-contacting transducers arranged to measure radial relative motion between the rotating shaft or element and the stationary element in the same transverse plane. Figure 6 provides a typical installation of such a non-contacting transducer system.

At each bearing, the transducers are normally mounted orthogonally in the bearing or as close to the bearing housing as possible. There could be occasions, such as on flexible shaft machinery, where it would be advisable to mount the transducers at locations other than at the bearings.

In any event, the transducer locations shall be arranged such that they provide adequate sensitivity to the machine dynamic forces. Although the transducers may be placed at any angular position, it is common to locate them vertically and horizontally or at $\pm 45^\circ$ from the vertical and horizontal, depending on ready access to the rotating shaft. A single transducer may be used if it is known to provide adequate information about the machine, however, some analysis techniques will then no longer be possible, such as orbit analysis and $S_{\max}$.



Key

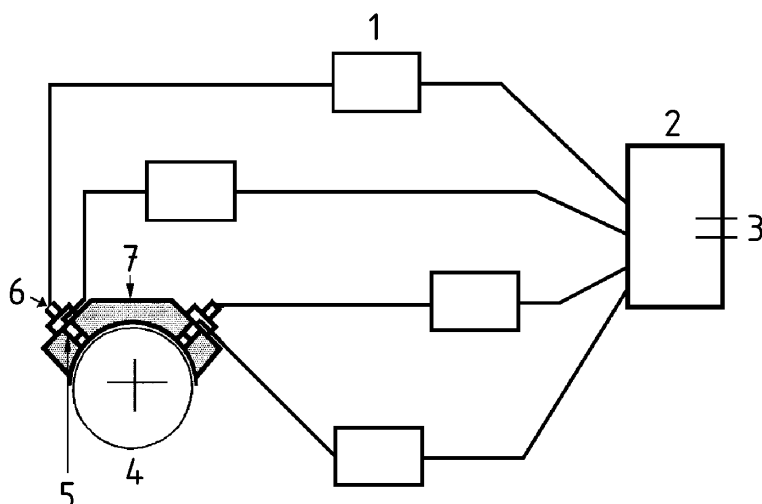
- 1 Local signal conditioners
- 2 To signal processing
- 3 Optional transducer orientations
- 4 Shaft
- 5 Non-contacting transducers

Figure 6 — Schematic diagram of relative motion measurement system using non-contacting transducers

5.2.6.3 Absolute motion of shaft

Some types of machines, such as those having soft rotor support structures and/or flexible rotors, or machines operating near a rotor natural frequency, could require the measurement of absolute motion of the shaft.

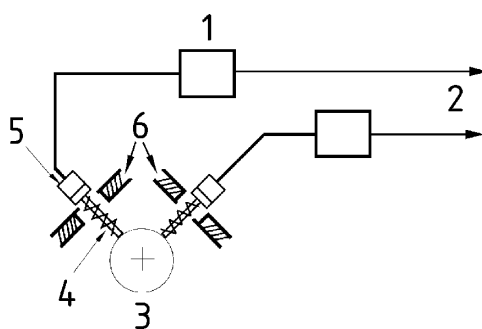
Figures 7 and 8 provide examples of measurement systems for determining the absolute motion of the shaft. The preferred method is a combination of non-contacting and seismic transducers, as shown in Figure 7. Although the shaft riding transducer shown in Figure 8 may be used in some cases, it should be noted that this method has a limited frequency range and that the method will not provide the shaft radial position.



Key

- | | | | |
|---|--|---|----------------------------|
| 1 | Local signal conditioners | 4 | Shaft |
| 2 | Remote readout instrumentation | 5 | Non-contacting transducers |
| 3 | Optional output for instruments for recording alarm/trips and/or data analysis | 6 | Seismic transducers |
| | | 7 | Machine structure |

Figure 7 — Schematic diagram of absolute/relative motion measurement system using a combination of non-contacting and seismic transducers



Key

- | | | | |
|---|---------------------------|---|---------------------|
| 1 | Local signal conditioners | 4 | Shaft riders |
| 2 | To signal processing | 5 | Seismic transducers |
| 3 | Shaft | 6 | Machine structure |

Figure 8 — Schematic diagram of absolute motion measurement system using shaft-rider mechanism with seismic transducers

5.2.7 Torsional vibration

Routine monitoring of torsional vibration on rotating machines is currently uncommon. It may become more frequently used in the future, in particular for monitoring variable-frequency a.c. driven machines. It has typically been used at a limited number of power plants in the electrical power generation industry, to gain improved understanding of the severity of interactions of the torsional dynamics of large steam turbine generator sets and the electrical oscillations in the transmission network. This involves monitoring turbine generator shaft torsional response due to transient disturbances in the network (e.g. faults, short circuits), and continuously acting steady-state stimuli from the network (e.g. unbalanced transmission lines). The monitoring equipment to do this is very specialized and can be very expensive, and it requires linkage to a computer for data processing to convert mechanical information acquired at several positions on the turbine generator shaft to provide estimates of fatigue life expenditure at the most critical positions within the machine (see also clause C.3).

When torsional vibration measurements are required, they are usually made by measuring the oscillatory speed variations of the machine at selected transducer locations. Commonly used transducer systems are as follows.

- Free-end mounted torsiograph which measures vibratory angular velocity. The vibratory velocity signal may be electronically integrated to provide a torsional displacement amplitude.
- Free-end or mid-shaft toothed wheels in combination with an electromagnetic transducer mounted on the stationary structure. The transducer provides a signal of the instantaneous oscillatory speed of the target “toothed” wheel. This signal is usually conditioned to provide a torsional displacement signal.
- Mid-shaft or free-end mounted disks with radial code markings in combination with optical transducers which measure the instantaneous speed and phase variations of the target lines. This signal is electronically conditioned to provide torsional displacement signals.
- Strain gauges mounted on the shaft to measure directly the alternating component of torsional strain.
- Laser systems.

Regardless of the type of measurement used, mathematical simulation is required to convert measurement information taken at a selected location to response estimates at other locations of interest. In general, the conversion of mechanical measurements to stress and fatigue responses is very complex, requiring sophisticated analytical procedures.

NOTE The hardware and software systems that have been used and the analysis of torsional vibration are well documented in the technical literature and will not be discussed in this part of ISO 13373.

5.2.8 Signal-conditioning systems

Signals derived from vibration transducers typically require some degree of conditioning to provide a useful measurement. Signal-conditioning functions may include conversion of the transducer output signal to a measurable quantity such as current or voltage, and processing of the signal into a suitable form for the measurement required.

Common examples of such condition processing are amplification/attenuation, filtering, scaling, d.c. offset and integration.

Depending on the complexity of the measurement system, the necessary signal-conditioning circuitry may be

- an integral part of the transducer,
- an integral part of the measurement instrument,
- separate, self-contained instrumentation in the signal path between the transducer and the measurement instrument, or
- a combination of the above.

In simple systems, where signal-conditioning functions are built into the transducer or the measurement instrument (or both), the user may have little or no choice as to signal-conditioner settings. However, in more complex equipment with broader variability and more options in signal conditioning, care shall be taken to maintain accurate records of all settings so that subsequent measurements duplicate the original settings. Comparison of measurements taken at signal-conditioner settings which are not identical can lead to very serious errors in condition assessment, because ensuing changes in measured quantities may be incorrectly attributed to changes in machinery condition.

When external signal conditioning is applied between the transducer and the measurement instrument, it is essential to be aware of signal levels and instrument dynamic ranges in order to avoid introducing distortion into the measurement.

Additionally, the frequency response characteristics of the signal conditioner(s) shall be properly matched to the remaining instrumentation to obtain valid results.

NOTE Information on signal processing and analysis will be given in ISO 13373-2, which is at present under preparation.

5.3 Measurement quantities

5.3.1 General

Vibration may be quantified in the form of linear or angular displacement, velocity or acceleration. Except in special cases (discussed in other clauses of this part of ISO 13373), the recommended quantity for condition monitoring on stationary parts of machinery is vibration velocity. For monitoring the relative position and motion of rotating parts, displacement is recommended. Acceleration is the recommended measurement quantity for condition monitoring of rolling element bearings and gears, for example, as they may exhibit faults at high frequency. In any case a selection of the measurement quantity should be based largely on the anticipated frequency of excitation.

Shafts with fluid-film bearing supports may experience a d.c. or static shift of the shaft position when the machine is placed in operation from rest. Although this displacement is not directly a vibration component, as there is no motion after the initial positioning, it is a value provided by the vibration displacement transducers, and it should be recorded since it provides the base position from which to measure the shaft dynamic vibration behaviour of the machine.

5.3.2 Magnitude range

The magnitude range to be measured shall be selected on the basis of previous experience or the criteria applied for evaluation of the particular machine being monitored, covering the lowest to the highest anticipated amplitude. In the absence of previous experience, refer to the applicable International Standard (e.g. the ISO 7919 or ISO 10816 series) for the magnitude range recommended for velocity measurements.

Test equipment should be designed to ensure that its self-noise is at least 10 dB below, or one-third of, the lowest vibration value to be measured, while ensuring that the system can accommodate signals at least 10 dB higher than the largest expected signals.

5.3.3 Frequency range

For reliable condition monitoring, measuring equipment shall be capable of covering a wide frequency range in order to encompass not only shaft rotational frequencies and harmonics, but also frequencies due to other components, such as bearings, gears, seals, blades or vanes. The frequency range measured should be tailored to the specific machine(s) being monitored, if possible, but it should normally not be greater than the maximum transducer linear range.

The maximum transducer linear range is the range of frequencies and amplitudes for which the calibration sensitivity of the transducer is a constant, within the specified measurement accuracy. Refer to 5.4 for further definition of measurement accuracy.

The linear frequency range of the system should generally cover frequencies from 0,2 times the lowest rotational frequency to 3,5 times the highest excitation frequency of interest (generally not exceeding 10 kHz). Typically, the

highest excitation frequency is the rotational frequency multiplied by the number of blades, gear teeth or vanes, etc., or it may be one of the rolling element bearing frequencies. For pumps, cavitation-excited vibration can also occur and be outside these ranges.

For mechanical suitability, 10 kHz is adequate. However, for diagnosing very high-frequency signals, as in the case of gears or rolling element bearings, the 10 kHz limit may be increased, even beyond the transducer manufacturer's recommended linear operating range, as shown in Figure 9. In these instances, although the individual high-frequency signals may not be completely accurate in amplitude, the signals can provide valuable information.

NOTE More information regarding very-high-frequency analysis will be given in ISO 13373-2, which is at present under preparation.

5.3.4 Phase

The phase angle between two vibration sources is an important consideration when evaluating signals. Phase is a measure of the angular or time difference of one sinusoidal vibration signal to another, or of a vibration signal to a fixed reference. For the purposes of condition monitoring, phase lag is commonly used. A fixed transducer, capable of generating a signal when a reference point on the shaft passes, is used as the phase reference. The phase lag corresponds, for example, to the delay time between a reference point on the shaft and the maximum or zero crossing of the vibration signal. The phase reference may also be used for synchronous time averaging.

The physical location of the phase reference point, often called a trigger location, is arbitrary. However, a keyway in the shaft, if accessible, is recommended as the reference point. Lacking a keyway, any other discontinuity of the shaft that exists only once per revolution and can create a distinct signal, may be used as a permanent phase reference.

The most common use of phase measurement is in the determination and correction of unbalances. In addition, phase measurements may be used to include fault detection through the measurement of relative motion between machine components, thermal vectoring, verification of misalignment, rotor cracks, the determination of cross-coupling effects and the identification of machine resonance.

When measuring phase between two locations, it is important to use similar transducers and associated signal-conditioning equipment in order to avoid instrumentation phase errors. Otherwise, compensations shall be made for the differences.

5.4 Measurement accuracy and repeatability

An acceptable measurement under this procedure falls into two categories as shown in Figure 9.

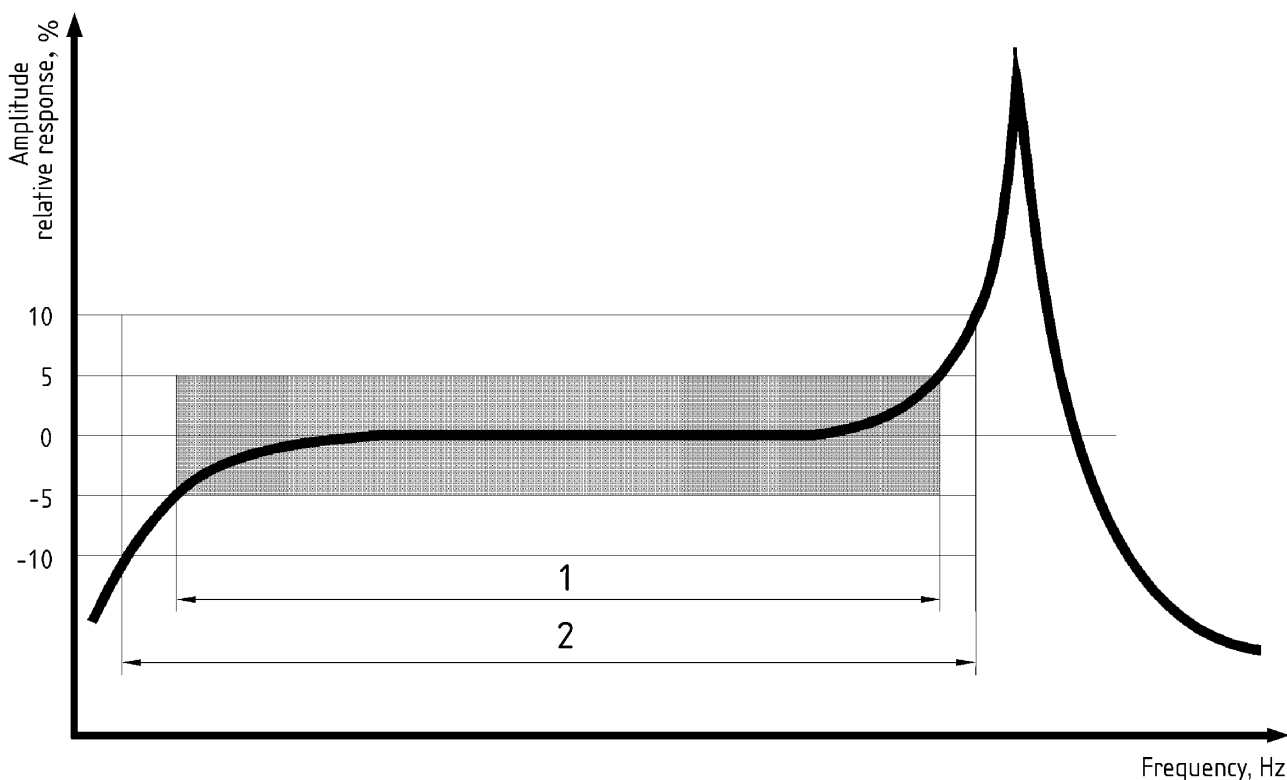
Type 1 measurements will have an allowable tolerance of $\pm 5\%$ of the calibration sensitivity for the required amplitude and frequency range of the measurement.

Type 2 measurements will have an allowable tolerance of $\pm 10\%$ of the calibration sensitivity for the required amplitude and frequency range of the measurement.

Measurements with greater than 10 % variations in calibration sensitivity over the required amplitude and frequency ranges are not in accordance with this procedure, unless special precautions are taken to return them to within the required tolerances. Measurements made in accordance with this procedure should be stated as such using the appropriate Type 1 or Type 2 designation.

Of equal importance, or in some cases more important, is the repeatability of the data. Therefore, the data should be taken with the same instrumentation, means of attachment, sensitivities and calibrations. Otherwise, unless accurately known compensations can be made, comparisons and trending of the machine condition signals can be misleading.

It should also be noted that the response curve of Figure 9 is of a general shape. Each transducer type described below has distinct characteristics, and the actual response curve should be obtained for each transducer selected.



- 1 Type 1 linear range
- 2 Type 2 linear range

Figure 9 — System frequency response

6 Transducers

6.1 Transducer types

There are two basic types of transducers for vibration condition monitoring, as follows:

- a) seismic devices that are normally mounted on the machine structure and whose output is a measure of the absolute vibration of the structure;
- b) relative displacement transducers that measure the vibratory displacement and the mean position between rotating and non-rotating elements of the machine.

There are some machinery measurements that require other types of transducers, such as strain gauges. However, their use for condition monitoring is less common.

6.2 Transducer selection

6.2.1 General considerations

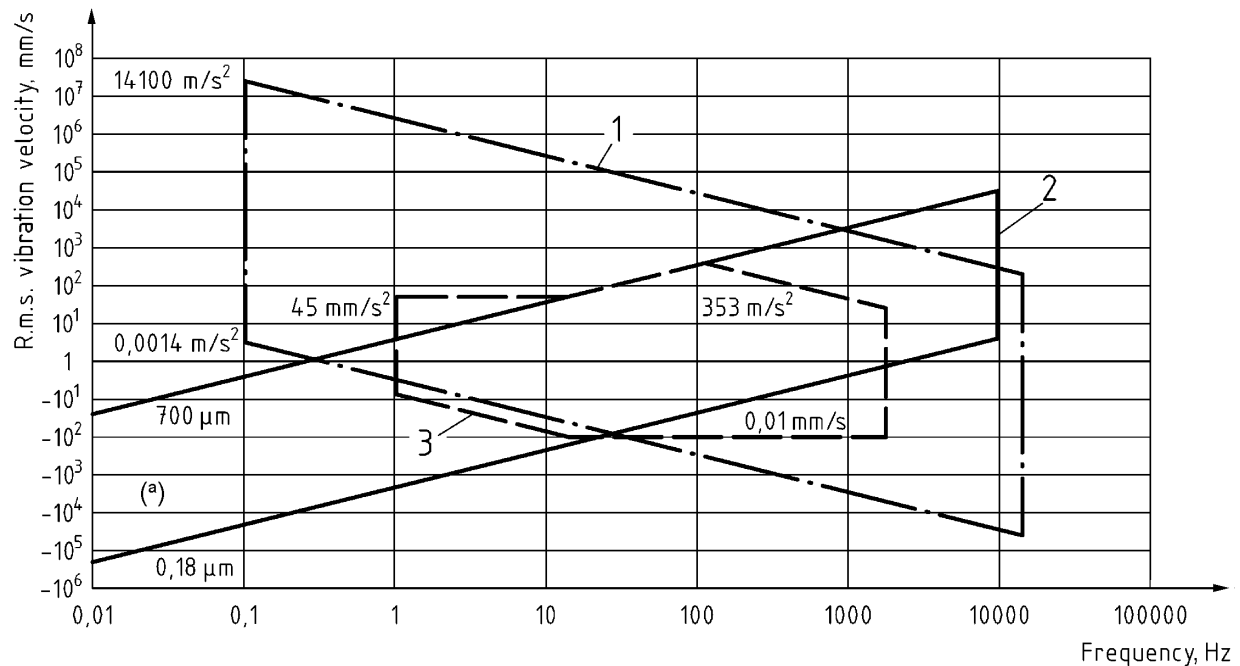
The selection of appropriate transducers depends on the particular application. In general, the transducers for condition monitoring are

- the accelerometer, whose output can be processed to yield any of the three parameters (acceleration, velocity and displacement),

- the velocity transducer, whose output can be integrated to yield displacement, and
- the non-contacting probe, whose output is directly proportional to the relative displacement between the rotating and non-rotating elements of the machine.

Figure 10 includes selection guidelines for the three types of transducers and their dynamic ranges versus frequency. The ranges shown include the majority of applications for condition monitoring. Under special circumstances, individual transducer ranges may be extended, primarily for diagnostic purposes.

NOTE The values given below are typical examples. Specific transducers can have different, especially broader ranges.



- 1 Piezo-electric accelerometer
- 2 Eddy-current proximity probe
- 3 Electro-mechanical velocity transducer
- a To d.c.

Figure 10 — Dynamic range versus frequency range for the application of transducers typically used for machine condition monitoring

6.2.2 Accelerometers

6.2.2.1 General

An accelerometer is a seismic device that generates an output signal proportional to the mechanical vibratory acceleration of the body being measured. In general, accelerometers are mounted on the stationary (non-rotating) structure of the machine. Accelerometers are available with various mounted resonant frequencies, typically from 1 kHz upwards. In general it is recommended that the linear range of the mounted accelerometer covers the frequencies of interest. It is common to integrate the output of an accelerometer to provide a velocity signal. However, caution shall be exercised when double-integrating to provide displacement, especially at low frequencies.

6.2.2.2 Typical range of application

The following are typical frequency, mass, amplitude and temperature ranges for accelerometers:

- frequency range 0,1 Hz to 30 kHz;
- for transducers used typically for condition monitoring, the mass ranges from 10 g to 200 g;
- special purpose accelerometers can measure a factor of 100 lower in amplitude than indicated in Figure 10;
- typical temperature range with internal charge amplifier is up to 125 °C;
- typical temperature range with external charge amplifier is up to 250 °C.

6.2.2.3 Peculiarities of usage

Accelerometers are available in a variety of sizes. They are characterized by a wide dynamic range (1 to 10^6) and a broad frequency range. They are used preferably in applications with small objects having little mounting area, at high ambient temperatures, or in strong magnetic fields. Disadvantages of external charge amplifiers are the sensitivities to insulation faults, large temperature fluctuations and movements of the connecting cables.

6.2.3 Velocity transducers

6.2.3.1 General

The velocity transducer is a device that generates a voltage signal directly proportional to the mechanical vibratory velocity of the body being measured. In general, velocity transducers are mounted on the stationary (non-rotating) structure of the machine. When displacement is the desired parameter, it is common to integrate the output of a velocity transducer to provide a displacement output.

The spring-mass characteristic of a velocity transducer will generally have a built-in resonant frequency of about 4 Hz to 20 Hz. At well below its natural frequency, the transducer output is proportional to jerk, which is the first derivative of acceleration, rather than velocity. In the frequency range near its natural frequency, the transducer output is dependent upon internal damping. In most cases, true velocity and phase proportionality are defined only well above the natural frequency of the transducer.

If it is necessary to use a velocity transducer below its natural frequency, it shall be suitably calibrated for the application. Modern electrodynamic velocity transducers with natural frequencies above 10 Hz have built-in conditioning electronics correcting the sensitivity in the frequency range below the natural frequency down to 1 Hz.

6.2.3.2 Typical range of application

The following are typical frequency, temperature and mass ranges for velocity transducers:

- frequency range 1 Hz to 2 000 Hz;
- typical temperature range is from –50 °C up to 200 °C;
- typical mass is 50 g to 200 g.

6.2.3.3 Peculiarities of usage

Velocity transducers are especially suited for the measurement of low-frequency vibration. The majority of velocity transducers used for monitoring have resonant frequencies above 10 Hz.

The advantage of velocity transducers is the relatively high output voltage at a low source resistance. The transducers are therefore relatively insensitive to insulation faults and electric fields. However, due to the presence

of internal moving parts, they can be sensitive to mechanical damage or wear if their rated operating range is exceeded. They are also very sensitive to high vibration in planes perpendicular to the measuring axis, and can give erroneous readings due to striction of the moving parts.

Electrodynamic transducers with a single coil are very sensitive to ambient magnetic fields and need good anti-magnetic shielding. Even with the shielding, when measurements are made on open electrical machines, serious interference can still be observed. Most electrodynamic transducers have at present two coils and are much less sensitive to ambient magnetic fields, allowing reduced anti-magnetic shielding and a lower mass.

6.2.4 Shaft displacement transducers

6.2.4.1 General

In rotating machinery, especially large critical turbomachinery, and machines where the supporting structure mass is high compared to the rotor mass, it may be necessary to measure the relative displacement between the rotor and stator structure. The proximity transducer is a non-contacting device which can directly measure the vibratory displacement and position of the rotating shaft relative to the stationary bearing or machine housing. The proximity transducer provides an a.c. component for vibratory motion and a d.c. component for position.

When the proximity transducer is used in combination with a seismic housing transducer whose signal is integrated to provide displacement, an absolute displacement measurement of the rotating shaft can be obtained by vectorially adding the two displacement signals. If the phase shift of the output signals from the seismic and proximity transducers are different, this shall be compensated for in the signal-conditioning equipment for the transducer combination.

6.2.4.2 Eddy-current principle

The most commonly used proximity transducers apply the eddy-current principle. A coil, carrying a high-frequency a.c. current, generates a high-frequency magnetic field. If electroconductive materials are introduced into this field (e.g. a machine shaft), eddy-currents are generated in this material, draining power from the high-frequency magnetic field. An oscillator circuit coupled to the coil generates a voltage which is proportional to the distance between the transducer and shaft. The oscillator may be internally housed within the transducer or may be an external part.

6.2.4.3 Typical range of application

The following are typical frequency, measurement and temperature ranges for displacement transducers:

- frequency range d.c. to 10 kHz;
- measuring range 1 mm to 10 mm;
- typical temperature range is from –50 °C up to 200 °C;
- typical temperature range, with internal oscillator, is from –50 °C up to 125 °C.

6.2.4.4 Peculiarities of usage

Generally, all parameters are influenced by ambient temperature variations. However, by using electronic compensation for thermal expansion, for example, this influence is normally kept within acceptable limits.

In addition, when using proximity transducers, the following precautions should be taken.

- a) The area around the probe tip shall be clear of conducting material.
- b) The measured area should be free of deposits of conducting material and there should be no discontinuities.

- c) When using different shaft materials, the proximity transducer shall be recalibrated.
- d) Non-homogeneous shaft material, shaft plating and residual magnetism produce a signal which is superimposed on the vibration signal (electrical runout). Surface irregularities of the shaft (out-of-roundness, grooves, etc.) shall be noted since they will also influence the measurement results (mechanical runout).
- e) Electrical runout is minimized when shaft surface property equalization techniques, such as shot-peening, micro-peening, rolling, induction hardening or abrasive techniques, are utilized.

6.3 Transducer attachment

6.3.1 General

Proper measurement of machinery vibration is critically dependent on accurately transmitting the motion to the transducer. The broadest range of fidelity is obtained with fixed transducer attachments. However, in some cases hand-held probes are sufficient.

For a complete description of transducer attachment methods of accelerometers and their effect on performance, refer to ISO 5348. General guidelines are included below.

The preferred method for attaching fixed transducers is a rigid mechanical fastening which is commonly achieved by drilled and tapped holes in the transducer and the machine, and joining the two by a threaded stud. Stud mounting has the ability to transfer high-frequency signals with little or no signal loss. The machine surface should be smooth, flat and clean. Also, application of a light coat of silicone grease, or equivalent, to all mating surfaces is recommended to improve the transmissibility and accuracy of the response signal, especially at high frequencies.

Where it is impractical or impossible to effect a stud-mounted mechanical connection, cements are used to fasten the transducer to the machine surface. The cement used shall be of the type that has high stiffness characteristics when cured. Resilient adhesives should not be used as they reduce the fidelity of the transmission of the signal.

Another common technique for non-intrusive transducer fastening is with a permanent magnet. However, it should be noted that the flatness of the mounting surfaces is critical in this technique. Both cement and magnetic methods could be subject to limits of frequency, temperature and amplitude, and therefore should be used with caution for condition monitoring.

6.3.2 Influence of transducer attachment

In some instances it may be necessary to mount the transducer(s) on a bracket which is, in turn, attached to the machine. In these cases, it is extremely important that all mechanical attachments be tightly secured. In addition, the linear range of the mounted transducer and bracket should cover the frequencies of interest.

Where permanently mounted transducers are impractical, hand-held probes are available. Hand-held probes are frequency-limited and are not normally recommended for use above 1 kHz. Both accuracy and repeatability are likely to be compromised by the use of hand-held probes. Moreover, some structural motions at the higher frequencies can invalidate hand-held probe measurements, even though such motions may not be detectable with the probe.

In order to demonstrate the effect on transducer performance of the various transducer attachment methods described above, the mounted resonant frequency of an accelerometer, with an internal 30 kHz resonant frequency, is typically reduced as shown in Table 1.

Velocity transducers are subject to the same reduction in performance. However, at present no International Standards exist that quantify the amount of degradation.

Table 1 — Effect of mounting on transducer performance

Mounting method	Effect on transducer performance (e.g. for a 30 kHz resonant transducer)
Rigid stud mount	No reduction in the resonant frequency of the accelerometer due to its mounting
Isolated stud mount	If a non-conducting rigid material, such as a mica washer, is introduced to prevent ground loops or other influences, the mounted resonance is slightly reduced to about 28 kHz
Stiff cement mount	The resonance is reduced to about 28 kHz
Soft epoxy mount	The resonance is reduced to about 8 kHz
Permanent magnet mount	The resonance is reduced to about 7 kHz
Hand-held	The resonance is reduced to about 2 kHz, but this method is not recommended for measurements above 1 kHz

7 Data presentation formats

7.1 General

Clause 5 provides information on the measurement quantities and types of measurements recommended for vibration condition monitoring. The data for vibration condition monitoring of machines are analysed and presented in many basic formats, which are initially established when taking baseline data. These formats include, but are not limited to

- trending the broadband values,
- frequency spectrum analysis,
- trending discrete-frequency spectral data,
- trending limited-frequency-band or narrow-band frequency spectral data,
- cascade (waterfall) analysis,
- Bode plots, Nyquist or polar plots, vector analysis, and
- shaft orbit analysis.

Other analysis and diagnostic methods, such as multivariate diagnostics and neural networks, are not covered in this part of ISO 13373.

NOTE More detailed information regarding the procedures for establishing the above data analysis techniques and data presentation and reporting formats will be given in ISO 13373-2, which is at present under preparation.

7.2 Baseline measurements

7.2.1 General

Baseline vibration data are those data or sets of data as measured or observed when the equipment operation is known to be acceptable and stable. All subsequent measurements will be compared to these baseline values to detect vibration changes. Baseline data shall accurately define the initial stable vibration condition of the machine, preferably operating in its normal operating mode and flow rate. For machines with several operational states, it can be necessary to establish baselines for each of these states.

For new and overhauled equipment, there can be a wear-in period. As a result, it is common to see a change in vibration during the first few days or weeks of operation. Therefore, time shall be allotted for wear-in before acquiring baseline data.

For equipment which has been operating for a significant period and monitored for the first time, a baseline can still be established as a trending reference point.

Vibration and operating data should be acquired at a sufficient number of intervals to establish that the machine has reached stable conditions. At this time, full baseline data should be taken and compared to applicable standards, as available, to determine the operability of the machine. The baseline signature should also be examined for evidence of undesirable conditions, such as shaft instabilities.

These data are the basis upon which future machine problems will be detected and diagnosed. These data shall be stored so that they are easily retrievable and secure.

Baseline vibration data can consist of all the potential vibration parameters that are commonly used to define the vibration condition of the machine. The more comprehensive the initial definition of baseline, the greater the likelihood of properly detecting, analysing and tracking the deterioration of the machine. The data used to define a vibration baseline for a condition monitoring programme may include all or some of the following:

- broadband magnitude (displacement, velocity and/or acceleration);
- time signal and waveform;
- rotational frequency;
- amplitude at once-per-revolution;
- vibration vectors (amplitude and phase);
- frequency spectrum analysis of the vibration signals at steady state;
- run-up/coast-down frequency response data (e.g. Bode plots, waterfall plots, polar plots);
- shaft orbit analysis;
- shaft centreline position.

The completeness of the baseline signature definition depends on the following:

- the importance of the machine;
- the previous history of the machine;
- the analysis equipment available;
- the capabilities of the personnel,
- other factors.

The locations at which data are obtained need not and should not be limited to those locations that are to be continuously monitored. It is recommended that the baseline be a comprehensive vibration analysis, normally encompassing more measurement points, directions, broader frequency range and finer resolution than are required in a routine programme. After either continuous or periodic monitoring of a relatively few points have established that a change has taken place, a repeat of the procedures used for baseline analysis can be prepared to help define the cause of the vibration change.

7.2.2 Broadband vibration

Unless prior knowledge dictates otherwise, the baseline broadband vibration should be acquired covering a sufficient dynamic range and frequency response to include all forcing excitations of interest within the machine. In general, this requires r.m.s. velocity 0,1 mm/s to 100 mm/s with frequency ranges of 0,2 times the lowest rotational frequency to 3,5 times the highest frequency of interest. For most machines, this frequency range will normally be

limited to 10 kHz on the upper end because of limitations of the measurement system. The lower frequency end is normally covered by a 10 Hz limit, with exceptions to be considered for low-speed machines.

Procedures for measuring broadband vibration of the rotor and the stationary structures can be found in the ISO 7919 and ISO 10816 series, respectively.

Baseline discrete-frequency vibration characteristics should be acquired using the dynamic range and frequency responses given above for broadband measurements, to confirm the potential causes of vibration excitations included in annex C. However, additional components may be evident. It is important to evaluate these anomalies which can lead to early detection of a problem.

Unexpected frequency components can also reveal some design configurations which are not included in annex C. These baseline data should be stored for ready reference for future machine evaluation and diagnostics.

7.3 Vibration trending

7.3.1 General

When setting up a trending programme, it is important to establish the time intervals of the measurements. On new or recently overhauled equipment operating under steady-state conditions, practical time intervals are dictated by the criticality of the machine, its reliability history and the data-storage capacity of the recording instrumentation. With continuous on-line systems, for example, if the vibration magnitudes do not change over a specified time, there is a preplanned “dumping” of old data that are considered to be of little value. For periodic systems, when the vibration starts to increase, it is advisable that the time intervals be reduced and surveillance increased. These guidelines are presented below.

When trending run-up or coast-down vibration, the measurements are taken at specified speed steps compatible with the type of machine. For large, relatively slowly accelerating machines, increments of speed as low as 5 r/min are sometimes used. However, this sample rate may not be practical for machines that have high rates of acceleration or deceleration. In these cases, continuous measurements during run-up or coast-down are advisable.

7.3.2 Broadband trending

7.3.2.1 General

There are three factors which shall be considered when establishing evaluation criteria for shaft or stationary structure vibration magnitudes, namely

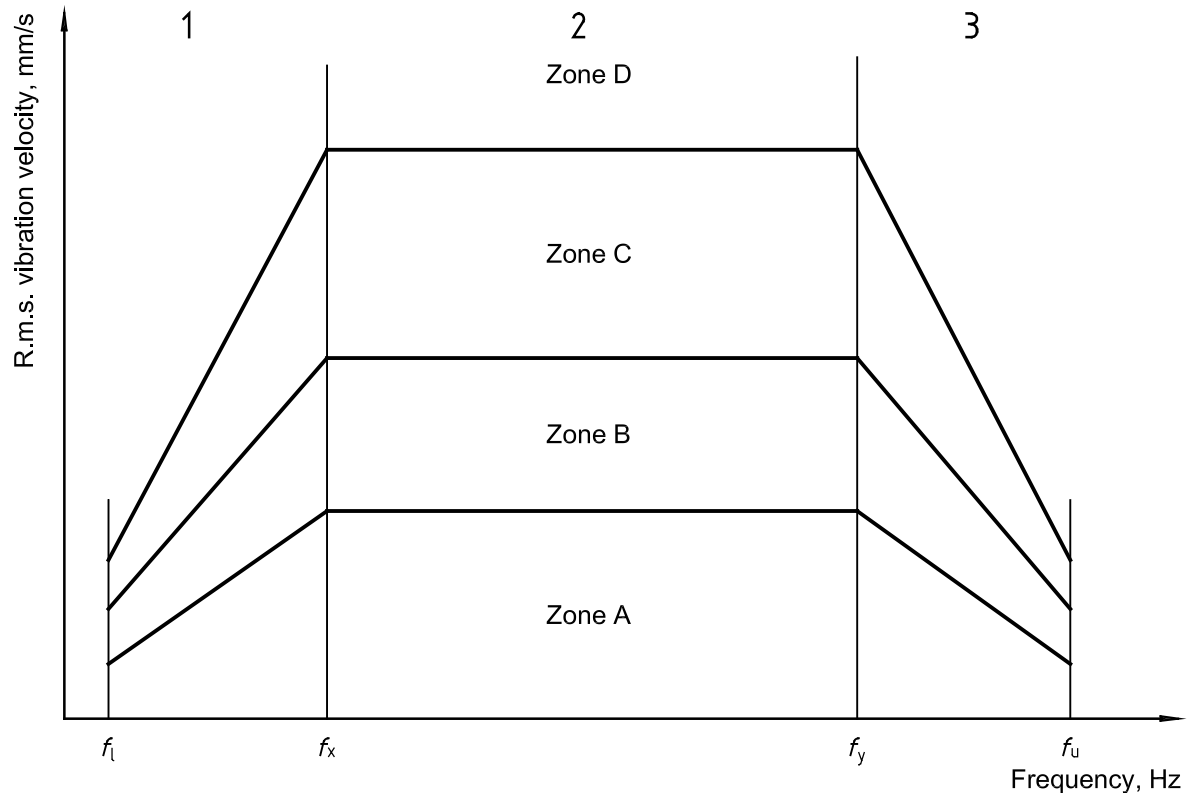
- a) vibration magnitude,
- b) any significant change in vibration, and
- c) the rate of change of the vibration.

When commissioning a new machine, the evaluations should be made on the basis of the vibration magnitude relative to some predetermined and agreed-upon permissible values. However, once a machine has been commissioned and normal operating magnitudes have been determined, any evaluation should be made not only on the basis of the magnitudes, but also on the basis of any significant change that takes place.

Two series of International Standards are available to provide criteria for determining when maintenance should be scheduled for rotating machinery. The first, ISO 7919, covers the measurement and evaluation of broadband vibration of non-reciprocating machines as measured on rotating shafts; the second, ISO 10816, covers the measurement and evaluation of broadband vibration as measured on non-rotating parts. Both series present evaluation criteria in terms of four zones and for changes in vibration.

In most cases, it has been found that vibration velocity is sufficient to characterize the severity of vibration over a wide range of machine operating speeds. However, it is recognized that the use of a single value for velocity, disregarding frequency, can lead to unacceptably large vibration displacements. This is particularly so for machines

with low operating speeds when the once-per-revolution vibration component is dominant. Similarly, constant velocity criteria for machines with high operating speeds, or with vibration at high frequencies generated by machine component parts can lead to unacceptable accelerations. Consequently, acceptance criteria based on velocity will take the general form of Figure 11. This indicates the upper and lower frequency limits f_l and f_h and shows that below a defined frequency f_x and above a defined frequency f_y , the allowable vibration velocity is a function of the vibration frequency. However, for vibration frequencies between f_x and f_y , a constant velocity criterion applies. The evaluation criteria and the values of f_l , f_h , f_x and f_y for specific machine types are given in the relevant parts of ISO 10816.



- 1 Constant displacement
- 2 Constant velocity
- 3 Constant acceleration

Figure 11 — Vibration zones

The vibration of newly commissioned machines would normally fall within zone A. Machines with vibration magnitudes within zone B are normally considered acceptable for unrestricted long-term operation.

Machines with vibration magnitudes within zone C are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in zone C until a suitable opportunity arises for remedial action. Vibration magnitudes within zone D are normally considered to be of sufficient severity to cause damage to the machine.

The ALARM values can vary considerably, up or down, for specific machines. The values chosen will normally be set relative to baseline magnitudes determined from experience for the measurement positions and directions for that particular machine or class of machines.

It is recommended that the ALARM value should be set higher than the baseline by an amount equal to 25 % of the upper limit of zone B. If the baseline is low, the ALARM may be below zone C.

In most cases where there is no established baseline, for example a new machine, the initial ALARM value should be based either on experience with other similar machines or relative to agreed acceptance values. After a period of time, the steady-state baseline values should be established and the ALARM settings adjusted accordingly.

In either case it is recommended that the ALARM value should not normally exceed 1,25 times the upper limit of zone B.

If the steady-state baseline changes (for example after a machine overhaul), the ALARM setting may need to be revised accordingly.

The TRIP values will generally relate to the mechanical integrity of the machine and will be dependent on specific design features which have been introduced to enable the machine to withstand abnormal dynamic forces. The values used will therefore generally be the same for all machines of similar design and would not normally be related to the steady-state baseline value used for setting ALARMS.

There may, however, be differences for machines of different design, and it is not possible to give clear guidelines for absolute TRIP magnitudes. In general, the TRIP magnitude will be within zone C or D, but is recommended that the TRIP value should not exceed 1,25 times the upper limit of zone C.

The zone boundaries recommended in the ISO 7919 and ISO 10816 series are based on statistical analysis of a worldwide industry survey.

The action required or recommended when equipment is operating in each of these zones depends on the rate at which the vibration magnitude is changing.

These same criteria should be used to specify the maximum period for obtaining data, or review of the data, obtained by continuous monitoring of equipment. The interval for monitoring or data review will vary depending on the magnitude of vibration observed and/or the rate at which the vibration magnitude is changing. These actions are described below and depicted in Figures 12 and 13.

7.3.2.2 Case 1: Equipment vibration magnitude is in the “normal range”

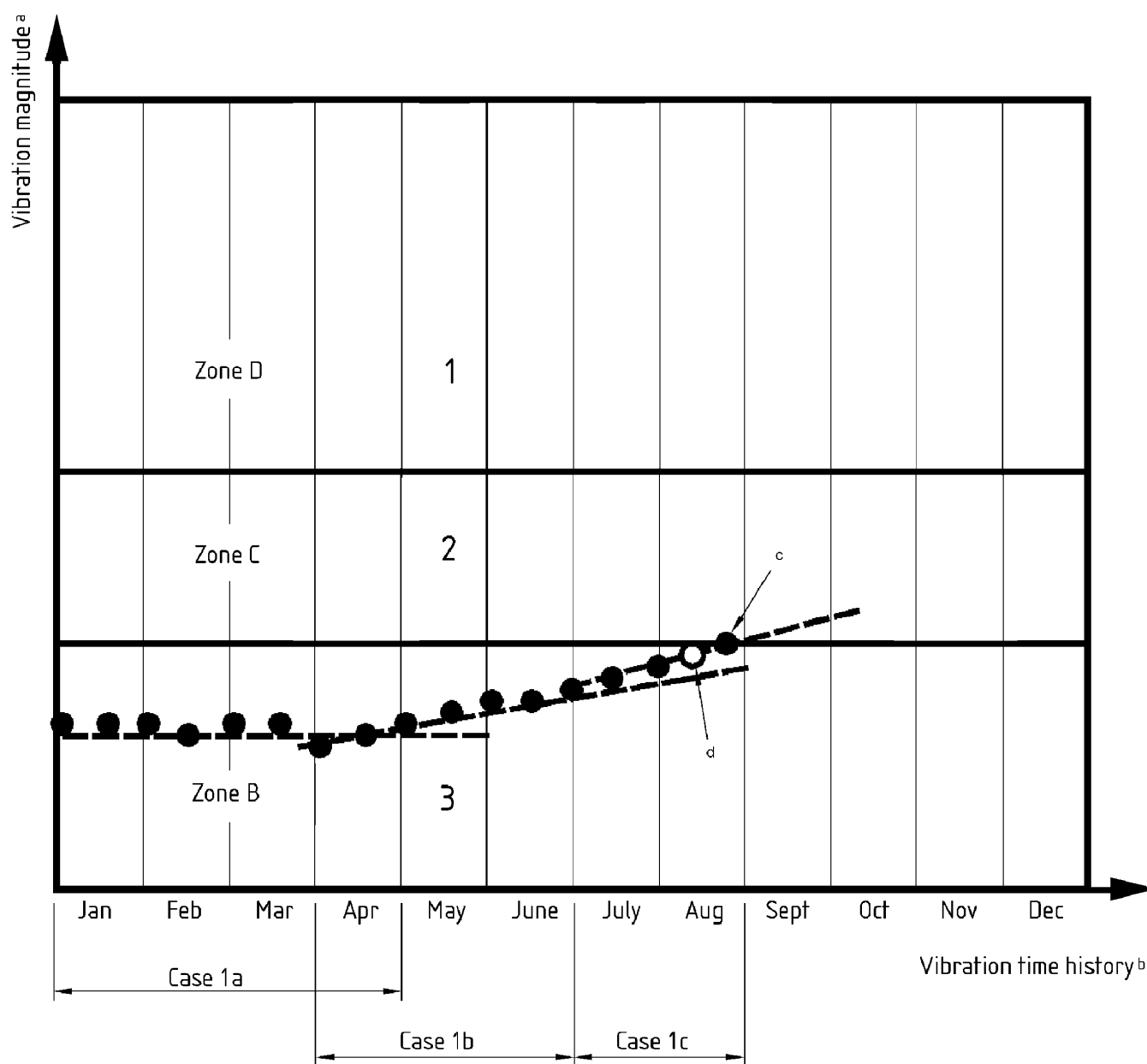
In this case the following action guidelines apply to the vibration trend curve in zone B of Figure 12.

- a) If there is no significant change in the magnitude from the previous data, then no action is required.
- b) If the vibration is increasing, the rate of increase is linear, and the magnitude is projected not to exceed the upper limit of the normal range before the next scheduled monitoring, no action is required. However, if the projected magnitude indicates that it will exceed that upper limit before the next monitoring, schedule more frequent monitoring as the upper limit is reached.
- c) If the rate of increase is non-linear, or the rate of change increases 25 % from a previous reading within a preset time frame, confirm the rate by continuous monitoring, or schedule more frequent monitoring and consider implementing a diagnostics programme.

7.3.2.3 Case 2: Current vibration magnitude is in the ALARM zone

In this case the following action guidelines apply to the vibration trend curve in zone C of Figure 13.

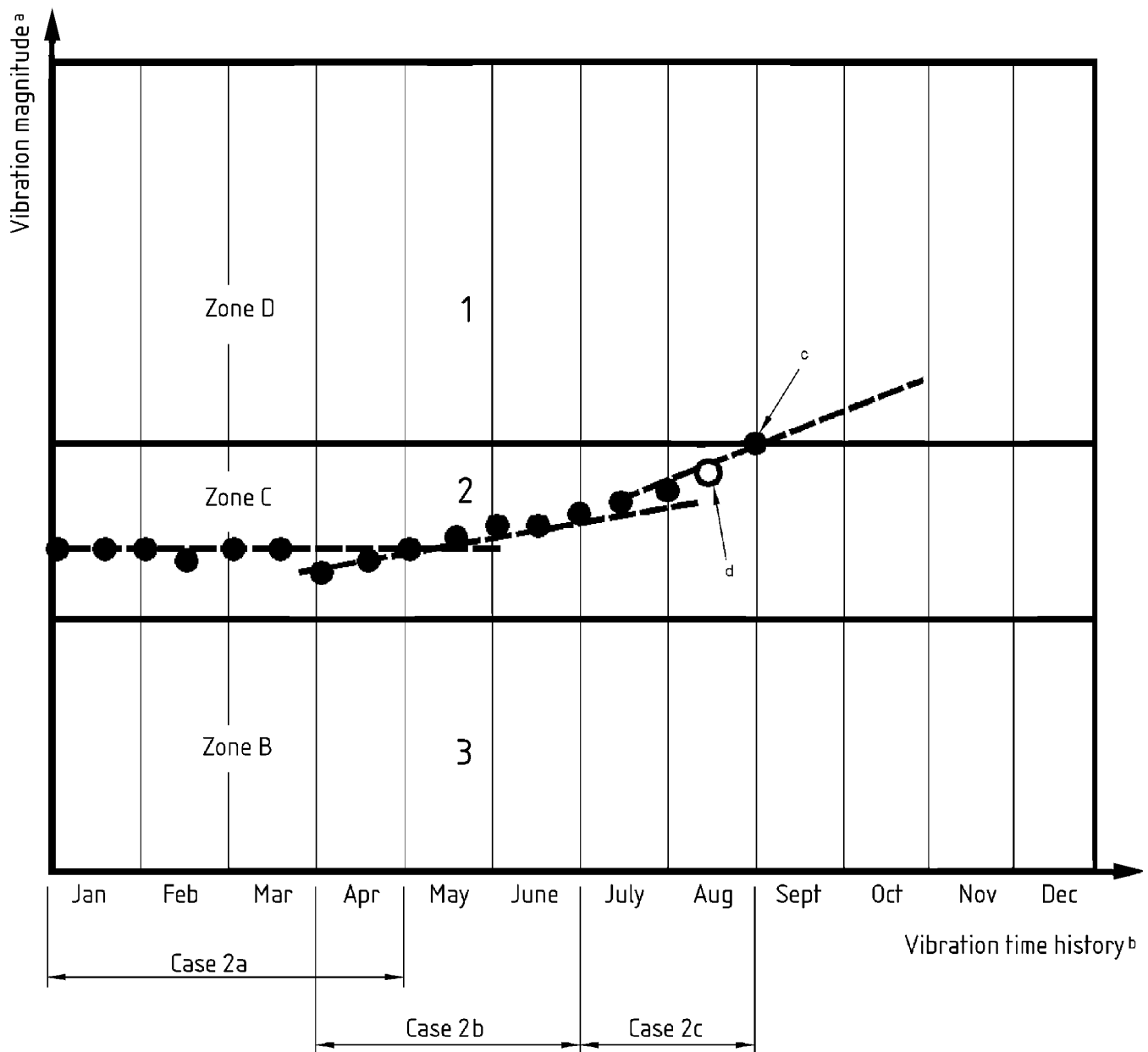
- a) If there is no change in the magnitude, maintain the same monitoring intervals.
- b) If the magnitude is increasing at a linear rate and is projected to exceed the action magnitude prior to scheduled maintenance, or before the next scheduled surveillance, or the rate of increase is non-linear, confirm the rate of increase by continuous or more frequent monitoring and reschedule maintenance. Increase the monitoring frequency to ensure that three data points are collected prior to rescheduled maintenance. A diagnostics programme is recommended to define the problem and maintenance required. Should a decrease in vibration magnitude be observed, biweekly monitoring rate should continue and diagnostics are recommended.



- 1 Action required
- 2 Alert
- 3 Normal

- a Peak-to-peak displacement (μm) or r.m.s. velocity (mm/s).
- b For defined vibration limits, see ISO 7919 or ISO 10816.
- c Projected occurrence of action required.
- d Follow-up required within 48 h.

Figure 12 — Broadband vibration as measured on rotating shaft or machine structure — Vibration magnitude in the “normal range”



- 1 Action required
- 2 Alert
- 3 Normal

- a Peak-to-peak displacement (μm) or r.m.s. velocity (mm/s).
- b For defined vibration limits, see ISO 7919 or ISO 10816.
- c Projected occurrence of action required.
- d Follow-up required within 48 h.

Figure 13 — Broadband vibration as measured on rotating shaft or machine structure — Vibration magnitude in the ALARM zone

7.3.3 Vibration during run-up/coast-down

Run-up/coast-down vibration as defined here is the vibration information obtained during the start-up and shut-down operations of a machine train. This type of data can provide insight into the mechanical condition of the machinery that cannot be obtained during steady-state operation. Unbalance response, the presence of structural and component resonance, including critical speeds, damping, electromagnetic anomalies, rubs and shaft cracks are all examples of conditions that are best detected and analysed from run-up/coast-down data. The spectra of

run-up/coast-down data contain a third dimension, either time or rotating speed. Thus, run-up/coast-down displays are more complex than equivalent steady-state spectra and, if plotted versus speed, require a once-per-revolution reference. The vibration data are usually displayed in presentation formats such as Bode plots, Nyquist/polar plots, Campbell diagrams and cascade (waterfall) diagrams.

Run-up/coast-down vibration is an important part of the baseline vibration data. The more comprehensive the initial testing, the greater the likelihood of properly detecting, diagnosing and tracking the deterioration of the machine.

7.3.4 Transient-vibration trending

Although transient displays are more complex than equivalent steady-state displays, they also have to be trended with time to detect the presence of changes in the machine. Comprehensive analysis of the complex three-dimensional data will detect anomalies which may not be disclosed in steady-state operation.

Even if the speed of a machine remains constant, transient operations can occur with variations in environmental conditions (temperature, pressure, etc.), load, process parameters, etc. Therefore, it is essential that the measurement of transient vibration associated with these changes be trended under the same operating conditions as far as possible.

7.4 Discrete-frequency vibration

7.4.1 General

Broadband vibration does not always provide sufficient information to identify the specific cause of an ALARM. This is especially true for complex equipment where several excitation frequencies appear in the frequency spectrum. In such cases it is advisable to split the broadband vibration signal into discrete frequency components (amplitude and/or phase). In most cases individual frequencies can be matched with corresponding moving machine parts. When these vibration components change, irregularities or machine damage can often be detected even at their earliest stage. Mechanically or thermally induced rotor unbalance, self-excited vibration, rubs, alignment changes, bearing or gear damage and rotor cracks are just a few of the types of problems that can be detected by discrete-frequency analysis.

In normal practices, a reference spectrum of the machine is recorded during commissioning or after overhaul, which becomes the baseline signature. This reference spectrum permits comparison with later frequency analyses in order to detect any change. Care should be taken to ensure that the same bandwidth and window function are used when comparing results from FFT analysis.

A sample frequency spectrum plot is given in Figure 14. It clearly shows vibration amplitudes at specific frequencies. It is important to evaluate the source of significant frequency peaks as their magnitudes can be abnormal, and when immediately investigated can lead to early detection of a problem. Unexpected frequency components can also reveal some design configuration which is not included in annex C.

NOTE More detailed procedures for converting a broadband time trace to a frequency spectrum will be given in ISO 13373-2, which is at present under preparation.

7.4.2 Trending of discrete-frequency vibration

Trending of the discrete frequencies within a machinery vibration spectrum provides more comprehensive data for establishing vibration limits. Figure 15 shows a typical example of the trending of the fundamental frequency and its harmonics as a function of time.

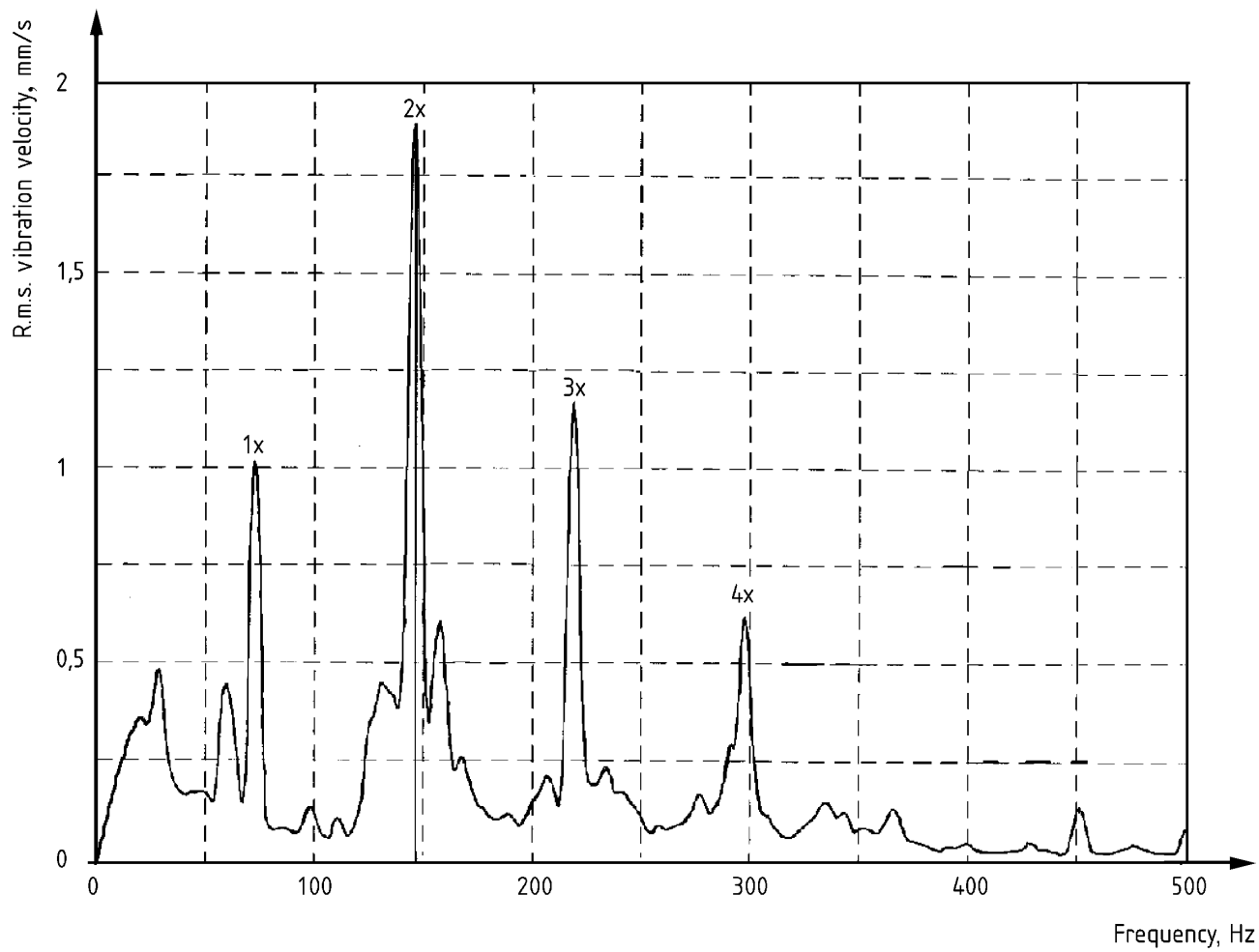
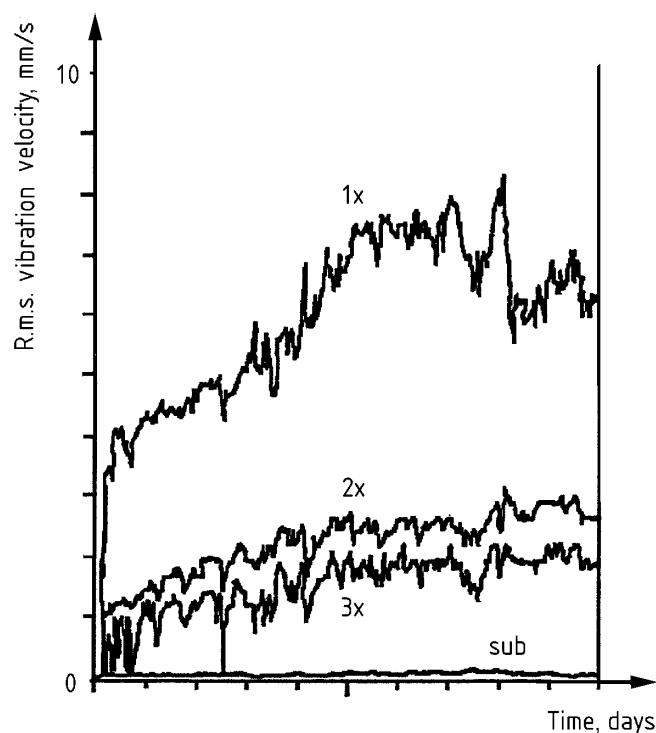


Figure 14 — Frequency spectrum



NOTE “sub” means subsynchronous components.

Figure 15 — Discrete-frequency vibration trends

7.5 Analysis of high-frequency vibration envelope

In some cases, changes in the machine vibratory state are more readily characterized by analysis of the high-frequency vibration envelope. There are a number of different techniques available but these are outside the scope of this part of ISO 13373.

8 Data analysis and communication

This part of ISO 13373 gives general guidelines to be followed for the acquisition of data for vibration condition monitoring of machines, in order to obtain meaningful data while also considering practical and economic constraints.

NOTE Further analysis and presentation formats of the vibration data will be given in ISO 13373-2, which is at present under preparation.

Annex A

(informative)

Guidelines for types and locations of measurements

Table A.1 consists of a listing of the various types of machines where vibration detectors are typically used to monitor the condition of a machine. Also included are the types of vibration transducers that are normally applied, together with guidance on their directions and locations. In many applications a reduced number of measurement locations may be used.

Identifications of the machines, transducers and locations are by no means limited to the information given in Table A.1; these are basic guidelines. However, for machine types not listed in Table A.1, it may be necessary to vary the transducer types and locations, as appropriate, for adequate vibration condition monitoring.

Table A.1 — Types and locations of measurement

Machine type	Evaluation parameters	Transducer type	Measurement locations	Direction	Standard reference	See note
Large steam turbine generator sets with fluid-film bearings Power generation	relative displacement or absolute displacement	non-contacting transducer	shaft at each bearing	radial $\pm 45^\circ$ or X and Y	ISO 7919-2	1
	velocity or acceleration	non-contacting and seismic transducer combination	each bearing housing	radial X and Y	ISO 10816-2	
	shaft axial displacement	velocity transducer or accelerometer	each bearing housing	axial Z		
	phase reference and speed	non-contacting transducer or axial probe	thrust collar	radial		
Medium and small industrial steam turbines with fluid-film bearings Mechanical drives, power generation	relative displacement	eddy current/inductive/optical transducer	shaft			
	velocity or acceleration	non-contacting transducer	shaft at each bearing	radial $\pm 45^\circ$ or X and Y	ISO 7919-3	
	shaft axial displacement	velocity transducer or accelerometer	each bearing housing and turbine housing	radial X and Y	ISO 10816-3	
	phase reference and speed	non-contacting transducer or axial probe	thrust collar	axial Z		
Small industrial steam turbines with rolling element bearings	velocity or acceleration	eddy current/inductive/optical transducer	shaft	radial		
	phase reference and speed	velocity transducer or accelerometer	each bearing housing and turbine housing	radial X and Y, axial Z	ISO 10816-3	
	relative or absolute displacement	eddy current/inductive/optical transducer	shaft	radial		
	velocity or acceleration	non-contacting transducer	each bearing housing	radial X and Y, axial Z	ISO 10816-4	
Large and medium industrial gas turbines with fluid-film bearings	velocity or acceleration	velocity transducer or accelerometer	shaft at each bearing	radial $\pm 45^\circ$	ISO 7919-4	
	shaft axial displacement	non-contacting transducer or axial probe	each bearing	radial X and Y	ISO 10816-4	
	phase reference and speed	eddy current/inductive/optical transducer	thrust collar	axial Z		
			shaft	radial		

Table A.1 (continued)

Machine type	Evaluation parameters	Transducer type	Measurement locations	Direction	Standard reference	See note	
Hydroelectric turbines with fluid-film bearings	relative displacement	non-contacting transducer	shaft at selected bearings	radial 90° apart	ISO 7919- 5	2	
	velocity or acceleration	velocity transducer or accelerometer	each bearing housing and turbine casing	radial 90° apart	ISO 10816-5	3	
	shaft axial displacement	non-contacting transducer or axial probe	thrust collar	axial Z			
	phase reference and speed	eddy current/inductive/optical transducer	shaft	radial			
Large pumps with fluid-film bearings - Boiler feed - Circulating - Process	relative or absolute displacement	non-contacting transducer	shaft at each bearing	radial ± 45°	ISO 7919-3, ISO 7919- 5	4	
	velocity or acceleration	velocity transducer or accelerometer	bearing housings	radial X and Y	ISO 10816-3, ISO 10816-5		
	shaft axial displacement	non-contacting transducer or axial probe	thrust collar	axial Z			
	phase reference and speed	eddy current/inductive/optical transducer	shaft	radial			
Medium and small pumps with fluid-film bearings	relative displacement (uncommon)	non-contacting transducer	shaft at each bearing	radial ± 45°	ISO 7919- 3	4	
	velocity or acceleration	velocity transducer or accelerometer	bearing housings	radial X and Y	ISO 10816- 3	5	
	shaft axial displacement	velocity non-contacting transducer or axial probe	thrust collar	axial Z			
	phase reference and speed	eddy current/inductive/optical transducer	shaft	radial			
Medium and small pumps with rolling element bearings	velocity or acceleration	velocity transducer or accelerometer	each bearing and pump housing	radial X and Y, axial Z	ISO 10816-3	4	
	phase reference and speed	eddy current/inductive/optical transducer	shaft	radial		5	
Vertically mounted pumps - Reactor coolant - Coolant pumps	relative displacement	non-contacting transducer	motor/pump shafts at each accessible bearing, top pump bearing (minimum)	radial 90° apart	ISO 7919- 5	3	
	velocity or acceleration	velocity transducer or accelerometer	motor and each accessible bearing housing	radial 90° apart	ISO 10816-3, ISO 10816-5		
	shaft axial displacement	non-contacting transducer or axial shaft probe	motor shaft	axial Z			
	phase reference and speed	eddy current/inductive/optical transducer	shaft	radial			

Table A.1 (continued)

Machine type	Evaluation parameters	Transducer type	Measurement locations	Direction	Standard reference	See note
Large electric motors and generators with fluid-film bearings	relative displacement	non-contacting transducer	shaft at each bearing	radial $\pm 45^\circ$	ISO 7919-3	
	velocity or acceleration	velocity transducer or accelerometer	housing	radial X and Y and axial Z	ISO 10816-3, ISO 10816-5	
	shaft axial displacement	non-contacting transducer or axial probe	thrust collar			
	phase reference and speed	eddy current/inductive/optical transducer	shaft	radial		
Medium and small motors with rolling element bearings	velocity or acceleration	velocity transducer or accelerometer	each bearing and motor housing	radial X and Y, axial Z	ISO 10816-3	5
Compressors – package centrifugal (four-poster), with fluid-film bearings and rigid housings	phase reference and speed	eddy current/inductive/optical transducer	shaft	radial		
	relative displacement	non-contacting transducer	each bearing, pinion housing and ball gear shaft	radial $\pm 45^\circ$	ISO 7919-3	
Compressors – centrifugal process	velocity or acceleration	velocity transducer or accelerometer	at each bearing or gear mesh	radial X and Y and radial	ISO 10816-3	6
	phase reference and speed	eddy current/inductive/optical transducer	each shaft			
	relative displacement	non-contacting transducer	shaft at each bearing	radial $\pm 45^\circ$	ISO 7919-3	
	velocity or acceleration	velocity transducer or accelerometer	bearing housings	radial X and Y and axial Z	ISO 10816-3	
Compressors – screw-type with two interlocking shafts, fluid-film bearings	shaft axial displacement	non-contacting transducer or axial probe	thrust collar or shaft end			
	phase reference and speed (if driving through gear box)	eddy current/inductive/optical transducer	shaft	radial		
	relative displacement	non-contacting transducer	shaft at each bearing	radial $\pm 45^\circ$		
	velocity or acceleration	velocity transducer or accelerometer	bearing caps and compressor housing	radial X and Y, axial Z	ISO 10816-1	
Compressors – screw-type with two interlocking shafts, fluid-film bearings	shaft axial displacement	non-contacting transducer or axial probe	thrust collar			
	phase angle of each shaft	phase reference transducer	each shaft	radial		

Table A.1 (continued)

Machine type	Evaluation parameters	Transducer type	Measurement locations	Direction	Standard reference	See note
Compressors – screw-type with two interlocking shafts, rolling element bearings	velocity or acceleration phase angle of each shaft	velocity transducer or accelerometer eddy current/inductive/optical transducer	bearing caps and compressor housing each shaft	radial X and Y, axial Z radial	ISO 10816-1	
Gears – large with fluid-film bearings	relative displacement velocity and/or acceleration shaft axial displacement	non-contacting transducer velocity transducer and/or accelerometer non-contacting transducer	shaft at each bearing bearing caps and gear case shaft ends	radial $\pm 45^\circ$ radial X and Y, axial Z axial Z		
Gears – with rolling element bearings	velocity and/or acceleration	velocity transducer and/or accelerometer	bearing caps and gear housing	radial X and Y, axial Z		
Fans – large with fluid-film bearings	relative displacement velocity or acceleration shaft axial displacement phase reference and speed	non-contacting transducer (uncommon) velocity transducer or accelerometer non-contacting transducer or axial probe eddy current/inductive/optical transducer	bearing housings casing bearing housings thrust collar or shaft end shaft	radial $\pm 45^\circ$ radial X and Y axial Z radial	ISO 7919-3 ISO 10816-3	7 8
Fans – medium and small with rolling element bearings	velocity/acceleration	velocity transducer or accelerometer	each bearing cap and fan housing	radial X and Y, axial Z	ISO 10816-3	
Centrifuges	relative displacement velocity or acceleration shaft axial displacement phase reference and speed	non-contacting transducer velocity transducer or accelerometer non-contacting transducer or axial probe eddy current/inductive/optical transducer	shaft at each bearing (if accessible) each accessible bearing thrust collar shaft	radial $\pm 45^\circ$ radial X and Y axial Z radial	ISO 10816-1	

Table A.1 (concluded)

Machine type	Evaluation parameters	Transducer type	Measurement locations	Direction	Standard reference	See note
Pulp refiners	relative displacement velocity or acceleration phase reference and speed	non-contacting transducer velocity transducer or accelerometer eddy current/inductive/optical transducer	shaft at each bearing bearing caps on each shaft	radial X and Y radial X and Y radial	ISO 10816-1	
NOTE 1 Orthogonal measurements at $\pm 45^\circ$ are normally used for condition monitoring. Vertical and horizontal (X and Y) locations are acceptable alternatives; that may provide more meaningful data particularly if these are the principal stiffness directions.						
NOTE 2 Measuring both the d.c. displacement and the dynamic displacements is always recommended practice. It is more important on hydroelectric turbines due to the relatively larger radial bearing clearances.						
NOTE 3 For vertical machines, measurements should be made in the axes of major and minor stiffness directions, usually related to the inlet location.						
NOTE 4 Pump vibration varies significantly with operating condition. Ideally, operation at the Best Efficiency Point (BEP) will have the least vibrations. Operating continuously at a point other than the BEP will produce increased vibrations. In any case, readings should always be taken under the same operating conditions for comparison.						
NOTE 5 Portable systems are commonly used as medium and small size equipment do not usually justify on-line systems.						
NOTE 6 If not directly coupled, two keyway references are required.						
NOTE 7 On large axial-flow fans, if the bearings and drive shafts are not accessible, measurements in the X and Y directions on the fan housing should be taken.						
NOTE 8 Overhung construction: main load-carrying bearing only.						

Annex B (informative)

Typical information to be recorded

B.1 Machine details

For each machine being monitored, the following information should be recorded:

- unique machine identifier (e.g. equipment code);
- machine type (e.g. motor, generator, turbine, compressor, pump, fan);
- power source (e.g. electric, steam, gas, reciprocating internal combustion (RIC), diesel, hydraulic);
- rated speed (e.g. r/min or Hz);
- rated power (e.g. kW);
- configuration (e.g. direct, belt or shaft driven);
- machine support (e.g. rigid or resiliently mounted);
- shaft coupling (e.g. rigid or flexible).

The following additional information may also be recorded:

- function (e.g. driver or driven).

B.2 Measurements

For each measuring system, the following information should be recorded:

- date and time (including time zone) of measurement;
- instrument type;
- transducer type¹⁾ (e.g. eddy current, velocity, accelerometer);
- transducer method of attachment (e.g. probe, magnet, stud, adhesive);
- measurement location¹⁾, orientation¹⁾ (e.g. description or code);
- measurement value (e.g. numeric quantity);
- measurement unit¹⁾ (e.g. μm or mm/s or m/s²);
- measurement units qualifier (e.g. peak, peak-to-peak, r.m.s.);
- measurement type (e.g. overall, amplitude over time, spectrum);
- FFT or other processing (e.g. filter, number of lines, number of averages).

¹⁾ Annex D gives examples of coding conventions for these items of information.

The following additional information may also be recorded:

- speed during measurement (e.g. r/min or Hz);
- power during measurement (e.g. kW);
- other significant operating parameters (e.g. temperature, pressure);
- calibration requirement, type and date of last or next required calibration.

B.3 Other information

Extra information on the machine and the measurements may be recorded in addition to the above, for example historical maintenance data.

Annex C (informative)

Potential causes of vibration excitations

C.1 Introduction

There are a wide variety of causes of machinery vibration. The causes usually result from deterioration and wear of the machine parts, abnormal operation, improper assembly, and maintenance or design deficiencies.

One common cause of vibration is unbalance in the machine rotor. This will produce vibration of the rotor as well as the stationary elements of the machine. The vibration will be most pronounced when the rotor rotational frequency becomes coincident with one of the machine natural frequencies. The vibration will be at rotor rotational frequency which is designated as $1\times$. The vibration can be reduced through balancing of the rotor to minimize the stimulus, which requires the addition or removal of mass from the rotor from specific locations. The phase of the vibration will change substantially as the rotor speed traverses each machine natural frequency. At a fixed speed, the vibration at any measurement point will usually be constant.

Another common cause of vibration in rotating machinery is shaft or bearing misalignment. There are two types of misalignment: internal and external. Internal misalignment occurs when the centrelines of components are not concentric. An example of internal misalignment is a non-uniform air gap in an electric motor. External misalignment is more common and occurs when the shaft centrelines of two or more machines coupled together are eccentric or angularly offset. Misalignment causes additional forces in a rotating machine which manifest themselves as vibration. Misalignment often produces (or changes) a one-time ($1\times$), two-times ($2\times$) or three-times ($3\times$) vibration component in the radial direction and, depending on the severity of the misalignment, even higher multiples (multiples of running speed frequency are designated as $2\times$, $3\times$, $4\times$, etc.).

C.2 Lateral vibration

Table C.1 includes these and other common causes of lateral machinery vibration related to rotational frequency and gives information on the expected vibration frequency characteristics and other comments to help evaluate a vibration problem. However, it has to be recognized that there are other frequencies to be considered.

At the outset of a machinery condition monitoring programme, it is recommended that all of the potential sources of vibration be considered, with associated operational and design specifics, to generate a table of vibration components to be expected. Typically, this table should include rotational frequencies and harmonics, oil whirl and whip, blade/vane passing, gear mesh, rolling element rates, etc. This table will be utilized in concert with the baseline discrete-frequency spectra (see 7.4), as well as the establishment of monitoring procedures, ALARM limits, data sampling time, diagnostics, etc.

It is important to recognize that analysis of the vibration frequency spectrum provides vital information to help evaluate vibration problems. Other determining factors include the amount of vibration phase shift due to speed changes or other causes, vibration amplitude stability and repeatability, direction of the principal vibration component relative to the machine rotor axis, and attention to locations of the machine where the vibration is most pronounced.

Table C.1 — Most common causes of machinery lateral vibration and resulting vibration characteristics

Cause	Characteristic vibration frequencies	Remarks (Phase measurements can give additional information for many causes.)
Unbalance	1× (i.e. once per revolution)	Changes in balance will give changes in the 1× vector. Vibration will be highest when running speed coincides with a rotor system critical speed. Significant vibration phase change will occur when passing through critical speeds. At a fixed speed vibration magnitudes are constant.
Bearing misalignment	1× or higher harmonics	Parallel or angular bearing misalignment is generally caused by foundation movements. Bearing misalignment is not a direct cause of vibration excitation but changes the dynamic characteristics of the support system.
Shaft misalignment	1×, 2× or higher harmonics	Angular/parallel misalignment due to coupling geometric inaccuracies. It introduces vibration excitation due to shaft bending. In some cases, the axial vibration component may be of similar magnitude to the radial components.
Journal bearing operating condition/ geometry	Subsynchronous or 1×, 2×, 3×	Changes in the bearing operating conditions or geometry can cause changes in the steady-state vibration at 1× and higher harmonics, or cause subsynchronous instability (oil or steam whirl). In the latter case the vibration is usually unsteady and can increase with time, often rapidly.
Rolling element bearing wear	Wideband acceleration at high frequency	Detection requires transducers with high-frequency response. Vibration tends to be localized to the region of the defective bearing. Vibration readings are usually unsteady and increase with time. Other techniques may be necessary to characterize the type of fault.
Stiffness dissymmetry (e.g. axial winding slots in generator/motor rotors)	2×	Vibration peaks when a 2× stimulus is coincident with a rotor critical speed. At a fixed rotor speed vibration magnitudes are constant. Compensating grooves are used on large machines to minimize this stimulus.
Bent rotor (see also thermal dissymmetry)	1×, 2× or higher harmonics	Change of 1× is most common. If the rotor is bent near the coupling, a high 2× axial vibration is frequently observed. At a fixed speed the rotor vibration values are constant.
Cracked rotor	1×, 2× or higher harmonics	A growth in the 2× vector is an indication that the growth of a transverse crack is getting critical. Changes in the 1× or higher harmonic vectors can also occur.
Component looseness in rotor	1× and harmonics of running speed frequency	Vibration values may be erratic and inconsistent between successive start-stop cycles. Sometimes subharmonic frequencies are also observed
Eccentric or non-circular journals	1× and for non-circular journals at harmonics of running speed frequency	Vibration values can be abnormal or excessive at low rotor speeds as well as at rotor critical speeds. At a fixed rotor speed the vibration values are constant.
Thermal dissymmetry	1×	Can be caused by non-uniform rotor ventilation or shorted electrical windings or non-uniform tightness of parts. Causes rotor to bow with the same vibration characteristics as for unbalance.
Gear defects	High frequencies corresponding to harmonics of gear mesh/rotational frequency and associated sidebands	Detection requires transducers with high-frequency responses. For defect in one tooth: 1× and multiples. For worn teeth: Gear mesh frequencies with sidebands and multiples.
Resonance	At excitation frequencies such as when rotor speed equals a natural frequency of the rotor/support system	Vibration magnification occurs at each machine resonant speed and large phase angle changes are evident in the 1× response as the rotor passes through critical speeds. Rotor unbalance is also the most common stimulus which can produce resonant responses of the machine in its non-rotating systems. On electric machines, the other major stimulus is at 2× which results from electromagnetic forces that the rotor induces on the stator.
Rubs	Most commonly 1×, but also multiples of 1×, subsynchronous frequencies and natural frequencies	Slight rubs that are initiated at low speed may clear themselves. However, rubs that are initiated at high speed may result in an abrupt change in vibration that rises rapidly to a magnitude that requires machine shut-down. Sometimes rubs occur due to machines being loaded too rapidly or as a result of sudden changes in the thermal condition within the machine. In other cases rubs may result from clearances being set too small between rotating and stationary parts, or a result of parts shifting during service.

C.3 Torsional vibration

For large steam turbine generator sets, there are a wide variety of planned and unplanned incidents that can cause electrical oscillations in the transmission equipment to which the generator is connected at its terminals. These incidents are described in Table C.2, as well as the resulting characteristic vibration frequencies experienced by the machine. In each case the incidents result in oscillating torques on the generator rotor, which in turn can stimulate shaft torsional vibration and vibration of other rotating and non-rotating components. The damping of torsional oscillations of a turbine generator is very low in comparison to damping of bending vibration. This is because for bending vibration significant damping is obtained through bearing journal radial motion that compresses the oil film, and this radial motion is not present in torsional vibration.

As shown in Table C.2, following transient disturbances in the network, the turbine-generator torsional response will be multimodal with a low decay rate. The frequency spectrum will generally show most response in the lower-order torsional modes, with some components at the electrical stimulus frequencies (generally the first and second harmonics of the electrical transmission system frequency).

There are also conditions in the electrical network which can cause a relatively low but continuously acting torsional stimulus from the generator rotor (e.g. untransposed transmission lines, unbalanced loads). Most commonly under these conditions the dominant frequency at which the generator rotor stimulates the shaft system is at twice the electrical system frequency. This frequency is high enough to stimulate relatively high-order turbine-generator torsional system modes. These modes are complex and need sophisticated vibration models to represent the possible bending vibration of low-pressure turbine blades coupled to the turbine rotor torsional vibration. The state-of-the-art of large steam turbine-generator design has made them robust to disturbances from the electrical network, so that torsional vibration monitoring is only rarely conducted.

Table C.2 — Most common causes of turbo machinery torsional vibration and resulting vibration characteristics

Cause	Characteristic vibration frequencies	Remarks
Electrical faults in transmission network including circuit breaker operation	At torsional natural frequencies and at multiples of electrical system frequency	Transient response. Shaft multimodal torsional response which is lightly damped
Generator internal electrical faults, terminal short circuits	At torsional natural frequencies and at multiples of electrical system frequency	Transient response. Shaft multimodal torsional response which is lightly damped
Mal-synchronization of generator to the electrical network	At torsional natural frequencies and at electrical system frequency	Transient response. Shaft multimodal torsional response which is lightly damped
Planned line switching incidents	At torsional natural frequencies and at multiples of electrical system frequency	Transient response. Shaft multimodal torsional response which is lightly damped
Fluctuating loads, e.g. generated by arc furnace near turbo generating plant	Responses at torsional natural frequencies	Transient response. Shaft multimodal torsional response which is lightly damped
Transmission system phase unbalance or untransposed lines	Forced steady-state response at twice electrical system frequency	Mechanical modes in the vicinity of twice system electrical frequency, may include coupling of shaft torsional dynamics and turbine blade bending vibration
Subsynchronous resonance	Unstable oscillations at a turbine-generator subsynchronous natural frequency	Due to electrical resonance in the electrical network (series capacitor compensated lines) or mis-operation of control equipment for d.c. transmission
Stick-slip in oil film bearing journals at low speed	Responses at torsional natural frequencies	Lightly damped multimodal response
Pulsating torques on synchronous electrical motors during run-up	At the torsional natural frequencies	Vibration will be highest during run-up as the rotor speed passes each torsional natural frequency
Stimulus from a.c.-d.c. converters that drive a.c. and d.c. motors	Responses at the harmonics of electrical system frequency	Steady-state forced response

Annex D **(informative)**

Conventions for identifying vibration measurement locations

D.1 Introduction

Recommendations for the measurement information to be recorded are given in annex B. Annex D is for information only and is based on the Machinery Information Management Open Systems Alliance (MIMOSA) conventions for identifying vibration measurement locations. MIMOSA is a convention for vibration transducer location, type and orientation which combines requirements from current specifications (e.g. API 670), common practice and a logical combination of the two that extends the convention to equipment where inconsistency appears to be the norm (e.g. vertical machines). It is cautioned that the user, if electing to adopt this convention, should ensure that the latest version of the MIMOSA convention is available.

During the process of developing the convention, a decision was made to include provisions for designating internal and auxiliary components such as shafts and accessory gears. A means is also provided to identify transducers, primarily for axial vibration, mounted on opposite ends of a machine such that machine motion in a given direction results in a positive signal from one and a negative signal from the other. These additions, explained in detail below, are not typically found in current schemes of measurement location identification. However, their necessity can be anticipated as automated expert diagnosis advances. Where the additions are not considered necessary or not included in current data, software translation routines should be constructed to accommodate the additions without placing any burden on the user. It is recognized that adding component identification and direction of motion increases the complexity of the measurement location convention. Flexibility to meet anticipated requirements without translation or a change in format was judged to be ample justification for the expanded convention.

D.2 MIMOSA convention

D.2.1 Definition

The convention defines location, transducer type and orientation separately. Six definitions are then combined into an unambiguous, fourteen character (no spaces), measurement location identification; see Table D.1.

EXAMPLE SFTA003AC090RN
(shaft A, bearing housing number 3, single-axis accelerometer positioned 90° counterclockwise from zero, mounted radially, normal motion).

The illustrative examples described below cannot cover all possible machine configurations. However, following the principles will assure full identification of any measurement on any machine.

D.2.2 Component part

Four user-defined alphanumeric characters provide a flexible means to identify specific component parts of a machine for convenience and purposes of automated diagnosis. Examples include the use of a paper machine (e.g. FRNT (front), BACK), a motor with shaft A (e.g. MTRA), a pump with shaft B (e.g. PMPB), a gearbox with shafts C and D (e.g. GBXC and GBXD), a specific roll (e.g. ROLL), and others where measurement identification requires additional amplifying information.

Table D.1 — MIMOSA convention rules

Definition	Length	Example
Component part (shaft, auxiliary gearbox, roll, etc.)	four alphanumeric characters	See D.2.2 XXXX if unknown
Location (housing number designation)	three digits	001 to 999
Transducer type code	two letters	See Table D.2 XX if unknown
Angular orientation	three digits	000 to 360, XXX if unknown
Transducer axis orientation	one letter	See Table D.3 X if unknown
Direction of motion	one letter	See D.2.7

D.2.3 Bearing housing location

D.2.3.1 General

A numeric sequence identifies the specific bearing housing on which a vibration measurement is recorded. Three numerals are sufficient for the vast majority of machines.

D.2.3.2 Horizontal process and turbo machines

Examples are turbines, motors, pumps, fans, compressors and associated speed changing gears.

The sequence begins with the bearing housing at the non-drive, uncoupled (outboard) end of the driver, designated number 001, and continues in numerical order towards the driven equipment to the last bearing housing on the first shaft axis. On multiple axis (geared) machines, the bearing sequence continues in numerical order from drive to driven along the second shaft axis, followed by the third until the end of the machine train is reached (see Figure D.1).

Bearing housings on gearbox idler or lay shafts follow the same numerical sequence across each shaft from drive to driven (see Figure D.2).

If two or more machines are driven from a single gear, bearing housings are numbered in sequence proceeding down the drive shaft to its end, then the sequence moves to the next shaft and proceeds to its end. This process repeats until all bearing housings are numbered (see Figure D.3).

When the driving machine is coupled at both ends, the bearing housing at the non-drive (uncoupled) end of the driven machine coupled to the thrust end (usually governor end on steam and gas turbines) of the driver is normally designated location number 001. The sequence continues as before (see Figure D.4).

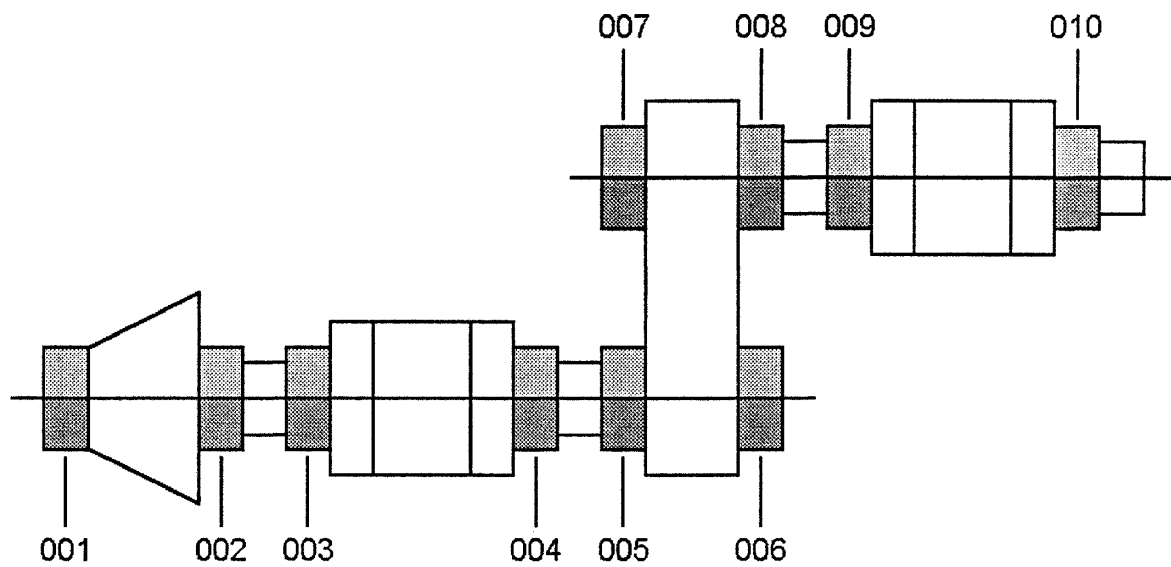


Figure D.1 — Bearing housing designation on a conventional process turbomachine

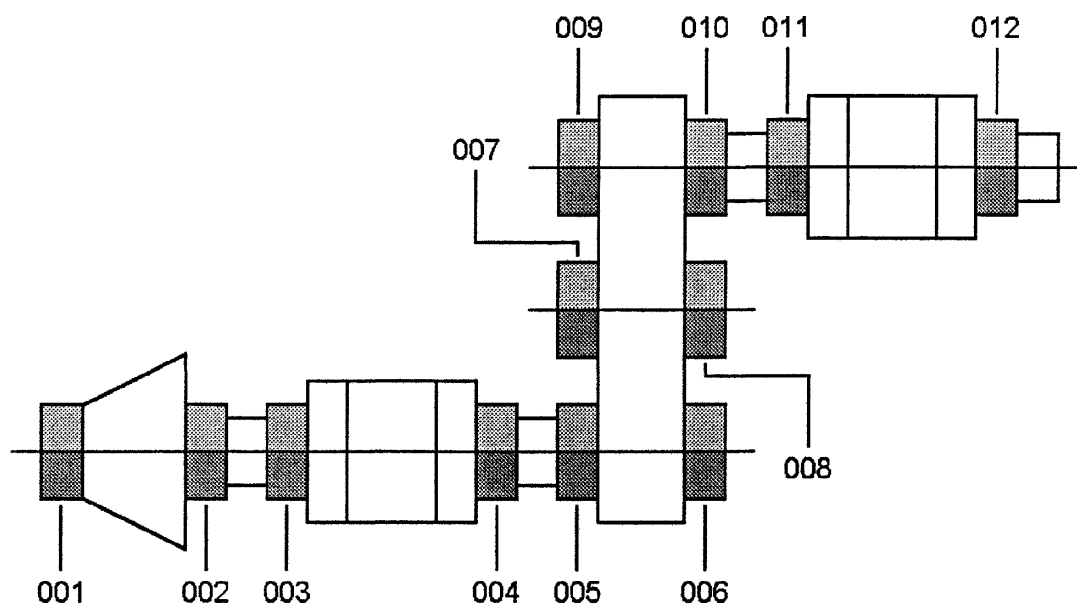


Figure D.2 — Bearing housing designation on gear with idler shaft

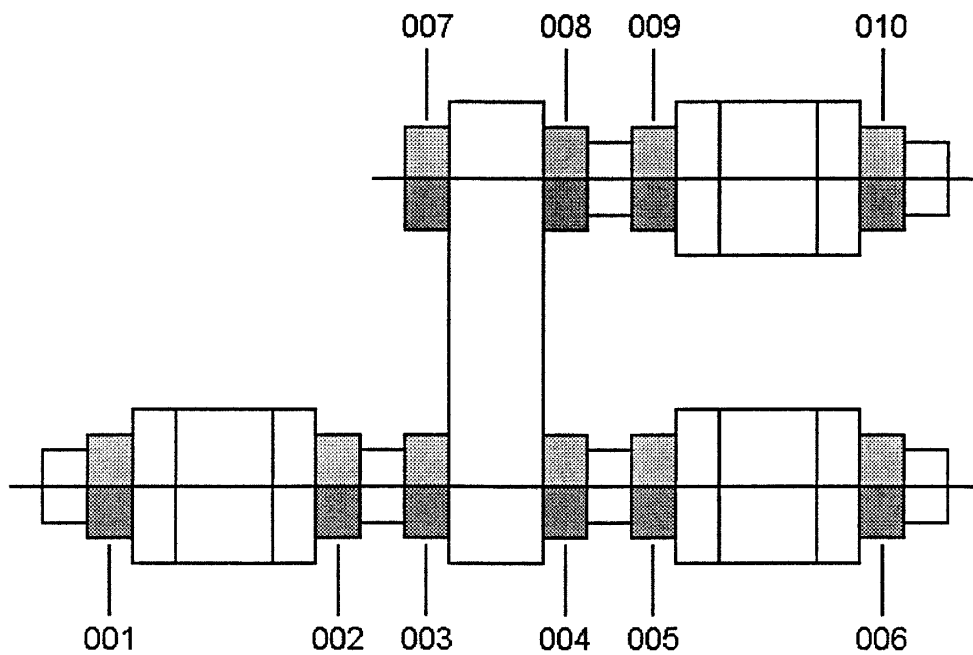


Figure D.3 — Bearing housing designation convention, gear with two or more driven shafts

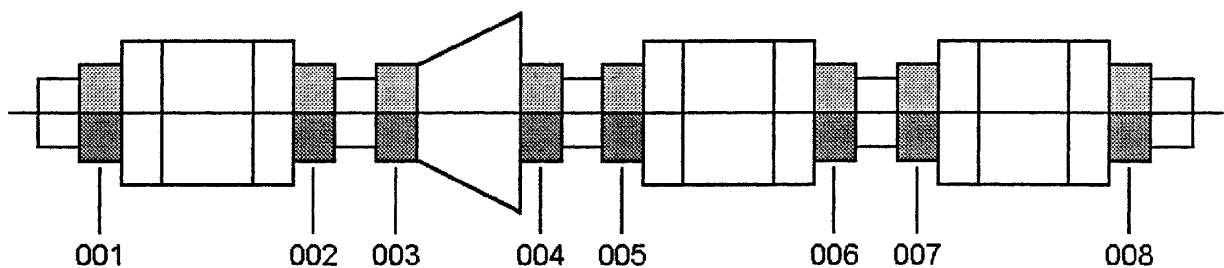


Figure D.4 — Bearing housing designation convention when both ends of the driver are coupled

D.2.3.3 Vertical machines

Vertical machines follow the same convention. The topmost bearing housing is designated 001, the numerical sequence proceeds down the shaft axis (see Figure D.6).

D.2.4 Transducer type

Transducer type is designated by a two-letter abbreviation according to Table D.2.

Table D.2 — Transducer type abbreviation listing

Code	Transducer type	Code	Transducer type
AC	Single-axis accelerometer	PD	Dynamic pressure
AV	Single-axis accelerometer with internal integration	PS	Static pressure
AT	Triaxial accelerometer	SG	Strain gauge
CR	Current	TC	Temperature-thermocouple
DP	Displacement probe	TR	Resistance temperature detector (RTD)
DR	Displacement probe used as a phase reference	TT	Torque transducer
LT	LVDT (linear voltage differential transformer)	TO	Torsional transducer
MP	Magnetic pick-up (shaft speed/phase reference)	VL	Velocity transducer
MI	Microphone	VT	Voltage
OP	Optical transducer	OT	Other

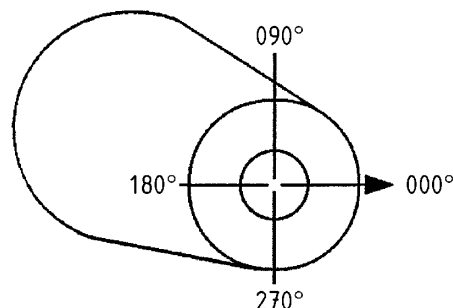
D.2.5 Angular orientation

D.2.5.1 General

The following identification for angular orientation is recommended as the best combination for existing conventions. In the 14-character identification, the degree sign is omitted.

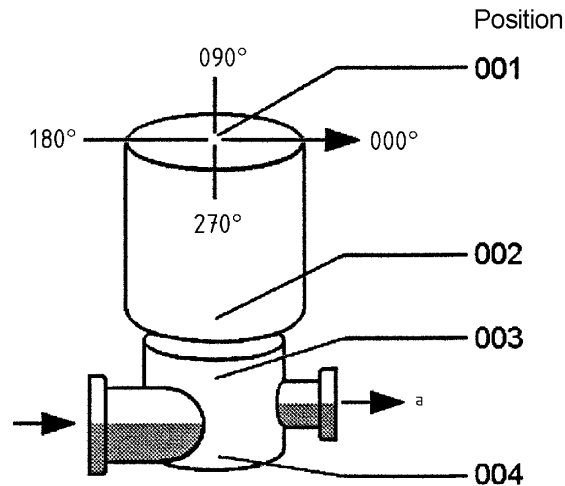
D.2.5.2 Horizontal machines

The angular position of a vibration transducer is measured from a zero reference located at 3 o'clock when viewed at position number 001, looking into the machine. The angle increases counterclockwise (regardless of the direction of shaft rotation) in the plane of shaft rotation from 0° to 360° (see Figure D.5).

**Figure D.5 — Angular convention, horizontal**

D.2.5.3 Vertical machines

The zero reference is located in the direction of flow with angular position measured counterclockwise in the plane of shaft rotation when viewed from the top (position 001) looking down (see Figure D.6). The zero reference on machines that reverse flow (e.g. pump storage units) is established for operation as a generator.



a Flow

Figure D.6 — Angular convention, vertical

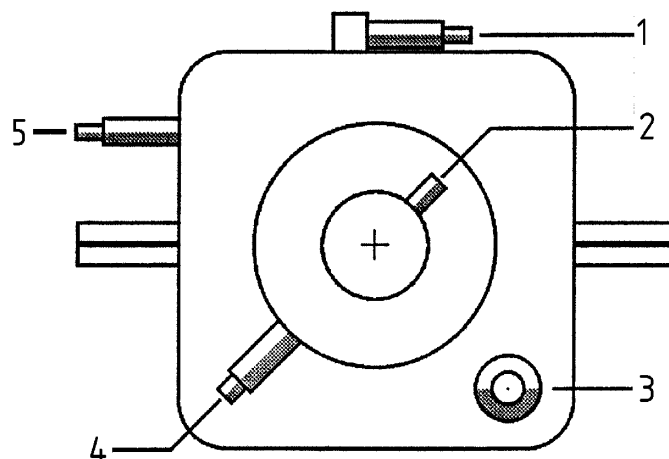
D.2.6 Transducer sensitive axis orientation

D.2.6.1 General

A single letter defines the direction of the transducer sensitive axis (see Table D.3). This portion of the identification provides unique descriptive information when the transducer sensitive axis does not coincide with the radial defined in D.2.5 (see Figure D.7: XXXAC135H, XXXAC090T, XXXAC315A). It is redundant when the sensitive axis coincides with the defined radial.

Table D.3 — Transducer axis orientation reference code

Code	Direction	Description
R	radial	transducer sensitive axis perpendicular to and passes through the shaft axis
A	axial	transducer sensitive axis parallel to the shaft axis
T	tangential	transducer sensitive axis perpendicular to a radial in the plane of shaft rotation
H	horizontal	transducer sensitive axis located at 000° or 180° only
V	vertical	transducer sensitive axis located at 090° or 270° only



Key

- 1 Tangential XXXAC090T
- 2 Radial displacement XXXDP045R
- 3 Axial XXXAC315A
- 4 Radial XXXAC225R
- 5 Horizontal, offset, XXXAC135H

Figure D.7 — Direction of transducer axis

D.2.6.2 Motion for a positive signal output

This convention requires the following criteria.

- a) Motion into the transducer is defined as positive (+), motion away from the transducer is designated negative (–) (see Figure D.8). This is the normal convention for casing accelerometers, velocity transducers and non-contacting shaft displacement (proximity) transducers.
- b) When radial transducers are installed in an X-Y pair, the X transducer will be 45° to the right (clockwise) from a radial bisecting the angle between the two transducers when viewed from position number 001 (regardless of the direction of the shaft rotation). The Y transducer will be 45° to the left (counterclockwise) from the bisecting radial (see Figure D.9). Utilizing this convention will always produce a correct orbital rotation.

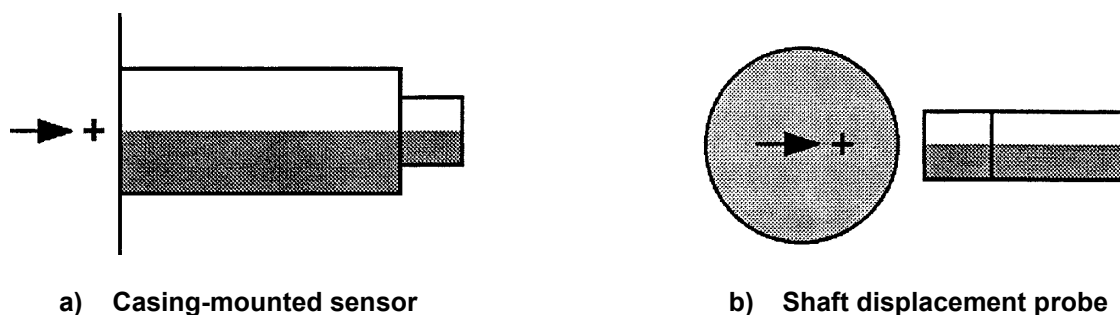


Figure D.8 — Motion conventions

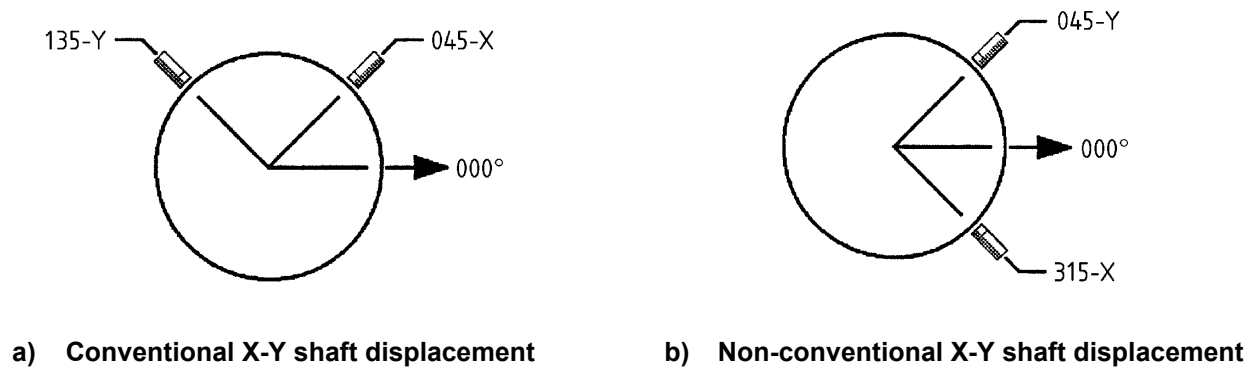
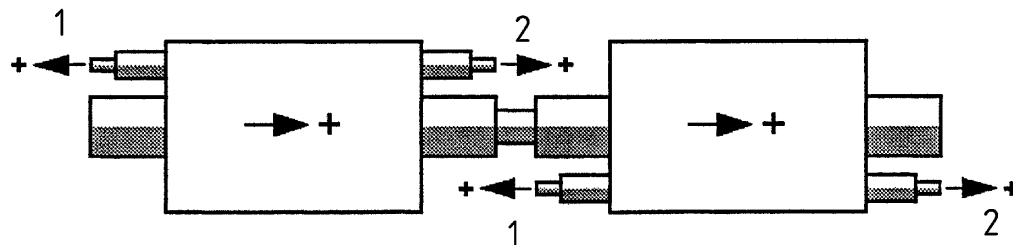


Figure D.9 — X-Y shaft displacement probe designs

D.2.7 Direction of motion

The final character in the measurement location identification code is either an N (normal) or R (reverse) to identify transducers mounted in opposition where machine motion in one direction results in positive motion in one transducer (N, normal) and negative motion (R, reverse) in the other. Axial transducers mounted in opposite directions at the two ends of a machine are the primary example (see Figure D.10). Axial machine motion towards the reference end is normally designated as positive. The axial transducer closest to the reference end of the machine, position 001, will be designated as normal (N) when mounted so that positive motion towards the transducer produces a positive signal output. Likewise, motion towards the reference end will produce a negative signal from the axial transducer at the opposite end, which is then designated R (reverse). The angular orientation defines the direction of motion for radially mounted transducers. Therefore, a default of N (normal) should be utilized for transducers mounted radially.



- 1 Reverse
- 2 Normal

Figure D.10 — Normal and reverse motion convention

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